Carrier system design manual

7 REFRIGERATION EQUIPMENT

IB

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reciprocating refrigeration machine

centrifugal refrigeration machine

absorption refrigeration machine

absorption-centrifugal combination

heat rejection equipment

SYSTEM DESIGN MANUAL

SUMMARY OF PART SEVEN

This part of the System Design Manual presents practical data and examples for selection and application of refrigeration equipment for normal air conditioning systems.

The text of this manual is offered as a general guide for the use of industry and of consulting engineers in designing systems. Judgment is required for application to specific installations, and Carrier is not responsible for any of the uses made of this text.

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CHAPTER 1. RECIPROCATING REFRIGERATION MACHINE

This chapter presents data to guide the engineer in the practical application of reciprocating refrigeration machines when used in conjunction with air conditioning systems.

The main component of these machines is the reciprocating compressor which is a positive displacement device employing the vapor compression cycle, and which is applied with refrigerants having low specific volumes and relatively high pressure characteristics.

A reciprocating refrigeration machine may be classified as one of the following:

1. A compressor unit consisting of a compressor, motor, and safety controls mounted as a unit.
2. A condensing unit consisting of a compressor unit plus an interconnected water-cooled or air-cooled condenser mounted as a unit.
3. A water-chilling unit consisting of either a compressor unit or a condensing unit, plus an interconnected water cooler and operating controls mounted as a unit.

Figures 1, 2 and 3 show a compressor unit, condensing unit, and water-chilling unit respectively.

TYPES OF COMPRESSORS

Compressors may be classified as either open or hermetic.

OPEN COMPRESSOR

An open compressor requires an external drive (*Fig. 1*) and may be direct driven thru a coupling or belt driven to operate at a specific speed, depending on load requirements. The type of the drive (electric motor, internal combustion engine or steam turbine) may be selected to provide sufficient horsepower to match the job requirement. This compressor may use any type of electric motor.

HERMETIC COMPRESSOR

A hermetic compressor has an electric motor and a compressor built into an integral housing (*Fig. 4*). The motor and compressor utilize a common shaft and bearings. The motor is generally cooled by suction gas passing thru the windings but may, in some cases, be water-cooled. These compressors eliminate problems of motor mounting, coupling alignment, motor lubrication, and refrigerant leakage at the shaft seal.

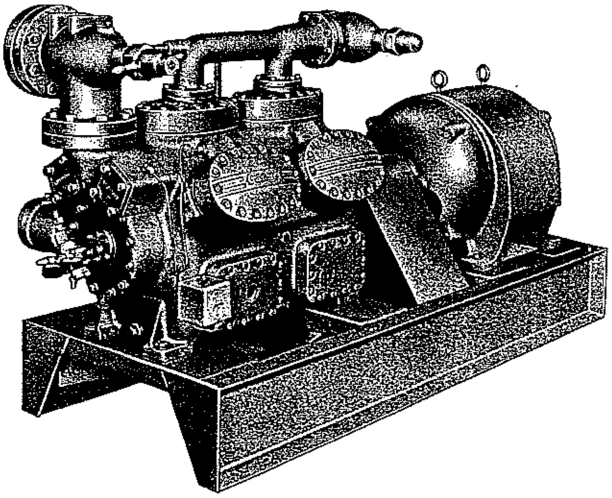


Fig. 1 — Open Compressor Unit

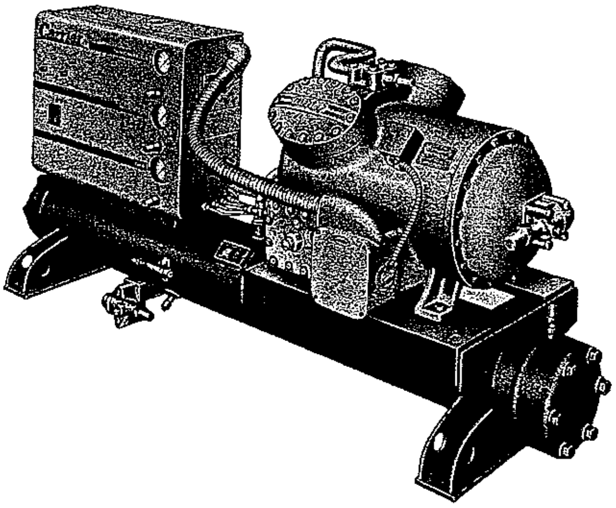


Fig. 2 — Hermetic Condensing Unit

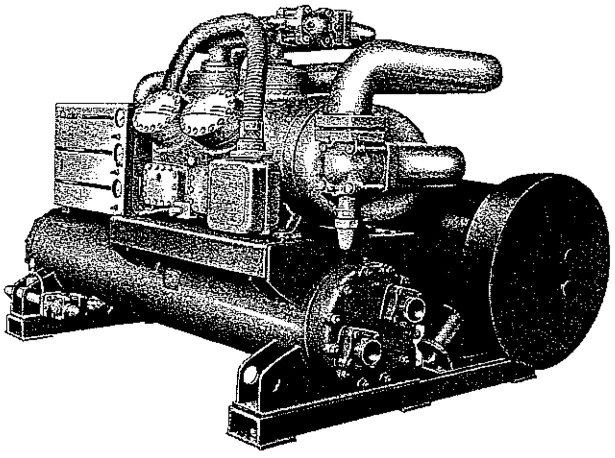


Fig. 3 — Water Chilling Unit

The compressor operating limits depend on the refrigerant used and the horsepower output of the motor. Generally, the motor horsepower is matched to the compressor and a refrigerant so that the motor does not overload when the unit operates within normal air conditioning levels. Hermetic compressors may be classified as either (1) sealed (requiring factory service for repairs) or (2) accessible (permitting service on the job).

APPLICATION

Refrigeration applications up to 60 tons generally utilize reciprocating compressors. From 60 to 200 tons either reciprocating compressors or other types such as centrifugal water chillers or absorption machines are used. Above 200 tons centrifugal water chillers or absorption machines are normally used.

A compressor unit must be combined with a device such as an air-cooled, water-cooled or evaporative condenser to condense the refrigerant. In a field built-up system this combination or a factory-assembled condensing unit can be applied with direct expansion evaporator coils in fan-coil equipment or a built-up apparatus. It may also be applied to a water or brine chiller or for any other type of refrigeration duty.

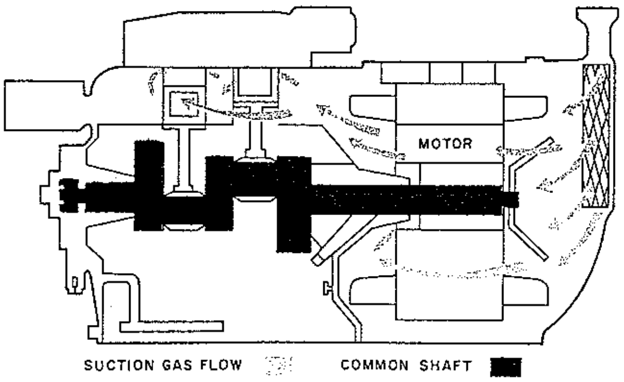


Fig. 4 — Cutaway View of Hermetic Compressor

A water-chilling unit may be applied to an air conditioning system or to any process requiring chilled water. Package water-chilling units can be obtained complete with or without a water-cooled condenser so that an air-cooled or evaporative condenser can be utilized if desired. When two units are required, they may be applied with their coolers piped in parallel or series water flow. Series connected coolers can be used on high rise water systems to effect a saving in the over-all horsepower per ton required for the system.

STANDARDS AND CODES

The location and installation of a reciprocating compressor should be in accordance with local and other code requirements.

The equipment should be manufactured to conform to the USAS B9 1 Saftey Code for Mechanical Refrigeration.

The cooler, condenser and accessories of the system should be built to conform to the ASME Unfired Pressure Vessel Code. This code covers the minimum construction requirements for design, fabrication, inspection and certification of unfired pressure vessels.

ARI Standards for open and sealed compressors establish recommended specifications for (1) standard equipment, (2) methods of testing and rating, including Standard Rating Conditions, and (3) provisions for safety. The Standard Rating Condition usually published by the manufacturer for a compressor unit used for air conditioning duty is Group IV which is based on an entering saturated refrigerant vapor temperature of 40 F, an actual entering refrigerant vapor temperature of 55 F, a leaving saturated refrigerant vapor temperature of 105 F, an ambient temperature of 90 F, and no liquid subcooling.

ARI Standards for a Reciprocating Liquid Chilling Package establish a Standard Rating Condition for a water-cooled model of a leaving chilled water temperature of 44 F, a chilled water range of 10 F, a .0005 fouling factor in the cooler and condenser, a leaving condenser water temperature of 95 F, and a condenser water rise of 10 degrees. The Standard Rating Condition for a condenserless model is a leaving chilled water temperature of 44 F, a chilled water range of 10 degrees, a .0005 fouling factor in the cooler, and a condensing temperature of 105 F or 120 F.

These Standard Rating Conditions can be used to make comparisons between compressors. When comparing catalog ratings of compressors of different manufacturers, the rating conditions must be known, particularly the amount of subcooling and superheating needed to produce the capacities shown.

Specifications should call for conformance to these standards and codes to assure a high quality product.

UNIT SELECTION

The selection of a reciprocating refrigeration machine is influenced by the economic aspects of the complete system; a balance between first cost and operating cost should be considered. The evaporator as well as the heat rejection equipment should be included in the analysis. Refer to *Chapter 5* for the economic consideration of the heat rejection equipment. Refer to *Part 6* for a discussion of dehumidifiers.

COMPRESSOR UNIT

Factors which influence the selection of a compressor unit include the following:

1. *Capacity* — The amount of heat to be exchanged by the refrigeration system in the evaporator. This heat depends on refrigerant weight flow and on entering and leaving enthalpy of the refrigerant at the evaporator.
2. *Evaporator Temperature* — The refrigerant temperature required to absorb heat from the medium being cooled. Lowering the evaporator temperature 10 degrees from a base of 40 F and 105 F reduces the capacity about 24%, and at the same time increases the compressor horsepower per ton about 18%.
3. *Condensing Temperature* — The refrigerant temperature required to reject heat to the condensing medium. Increasing the condensing temperature 15 degrees from a base of 40 F and 105 F reduces the capacity about 13%, and at the same time increases the compressor horsepower per ton about 27%.
4. *Refrigerant* — The four main refrigerants used in reciprocating compressors are Refrigerants 12, 22, 500 and 502. *Table 1* indicates some comparative data on these four; refer to *Part 4* for more information.
5. *Subcooling of the Condensed Refrigerant* — Subcooling increases the potential refrigeration effect by reducing the percentage of liquid flashed during expansion. Subcooling may be accomplished in the condenser, in an external subcooler, or in a liquid suction heat exchanger. For each degree of subcooling, the compressor capacity is increased about 0.5% due to the increased refrigeration effect per pound of refrigerant flow.
6. *Superheating of the Suction Gas* — Superheating can occur by heat pickup in piping outside the cooled space, in a liquid suction heat exchanger, or in an evaporator within the cooled space. Superheating increases the compressor capacity slightly (0.3-1.0% per 10 degrees) when using Refrigerants 12, 500 and 502, providing the heat absorbed by the vapor represents useful refrigeration such as coil superheat, not superheating from a liquid suction heat exchanger. Superheating from a liquid suction heat exchanger increases the capacity of the compressor by the subcooling effect on the condensed liquid. Compressor ratings are generally published with a maximum actual suction gas temperature of 65 F

TABLE 1—COMPARATIVE DATA OF REFRIGERANTS

Refrigerant No. (ARI Designation)	12	22	500	502
Chemical Name	Dichlorodifluoromethane	Monochlorodifluoromethane	Azeotrope of Dichlorodifluoromethane and Difluoroethane	Azeotrope of Monochlorodifluoromethane and Monochoropentafluoroethane
Chemical Formula	CCl_2F_2	$CHClF_2$	73.8% CCl_2F_2 and 26.2% CH_3CHF_2	48.8% $CHClF_2$ 51.2% $CClF_2CF_3$
Boiling Point at 1 atmosphere (F)	−21.62	−41.4	−28.0	−50.1
Saturation Pressure (psig) at: 40 F 105 F	51.67 141.25	83.72 227.65	60.94 167.85	94.90 244.40
Net Refrigerating Effect (Btu/lb) 40 F to 105 F (no subcooling)	49.13	66.44	59.82	43.72
Displacement (cfm/ton), 40 F to 105 F (no subcooling, no superheating)	3.14	1.98	2.69	2.04

for Refrigerants 12 and 500 and 75 F for Refrigerant 502.

Although tests using Refrigerant 22 in a compressor indicate a negligible increase in capacity due to improvement in the volumetric efficiency, superheating is not recommended because of the possibility of compressor overheating. Therefore, compressor ratings for Refrigerant 22 are generally published with only the superheat obtained with the normal expansion valve action and line transmission losses. Superheat is generally limited to 15 to 20 degrees between 40 F and 20 F saturated refrigerant temperatures respectively. Liquid suction heat exchangers are not used with Refrigerant 22 except where necessary to evaporate liquid refrigerant in the oil returning to the compressor and to eliminate liquid slugging to the compressor.

7. *Refrigerant Line Pressure Drops* — The operation of the reciprocating compressor in a refrigeration system is similar to that of a pump in a water system. It must be selected to overcome the system resistance and to produce the required refrigerant flow. At normal air conditioning levels, a piping loss equivalent to approximately 2 degrees is allowed in the suction piping, and a loss equivalent to 2 degrees is allowed in the hot gas discharge piping. Thus, when an evaporator requires a refrigerant temperature of 42 F to handle a required load, the compressor must be selected for a 40 F suction temperature (42 – 2 piping loss). Correspondingly, if the condenser requires 103 F to reject the proper amount of heat, the compressor must be selected for a 105 F condensing temperature (103 + 2 piping loss).

   At lower suction temperatures it may be necessary to allow a larger difference between the evaporator and compressor suction temperatures because of the pressure-temperature relationship of the refrigerant. For example, with Refrigerant 12 a pressure change of 1 psi at 0 F is equivalent to a temperature change of 2 degrees, and at +50 F equivalent to 1 degree. For Refrigerant 22 a change of 1 psi at 0 F equals a change of 1¼ degrees, and at +50 F equals ⅔ degree. For Refrigerant 500 a change of 1 psi at 0 F equals a change of 1⅔ degrees, and at +50 F equals ¾ degree.

8. *Operating Limits* — The manufacturer of a compressor unit generally states the operating limits of his compressor. Capacity limits may be shown in the rating tables with a note stating that extrapolation of the ratings is not permitted. There may be limitations on suction temperature, superheat, compression ratio, discharge temperature, compressor speed, horsepower, or motor cooling. As examples of these limitations, most manufacturers limit (1) the saturated suction temperature to a maximum of 50 F, (2) the compression ratio for Refrigerant 22 compressors to 5, and (3) the discharge temperature at the discharge valve to 275 F. This compression ratio limitation may be exceeded if water-cooled heads are used. The horsepower or kilowatt input may be limited by the size of motor available with the compressor. Generally, this limitation occurs with hermetic compressors which have built-in motors of specific sizes. If 50-cycle motors are used, the compressor speed is reduced and the capacity has to be adjusted accordingly. For all practical purposes the capacity is approximately proportional to the speed.

9. *Heat Rejection* — In order to select a condenser to match the compressor, the heat rejection of the compressor must be known. This is usually given in the ratings by the manufacturer, or may be approximated by multiplying the capacity by a given heat rejection factor. The heat rejection is dependent upon the compressor brake horsepower for the conditions involved, less the heat transferred to ambient air, jacket water or oil cooler during the compression of the refrigerant vapor. For an open type compressor the heat rejection can be approximated by adding the required brake horsepower converted to tons to the refrigeration capacity. (To convert brake horsepower to tons, multiply by 0.212.) For a gas-cooled hermetic type compressor the heat rejection can be approximated by adding the kilowatt input in tons (or Btu) to the refrigeration capacity. (To convert kilowatts to tons, multiply by 0.285.)

A typical rating of an open type compressor unit is shown in *Table 2*.

Example 1 — Selection of a Compressor Unit

Given:
  Evaporator conditions
    Load = 36 tons
    Evaporator temperature = 30 F
    Coil superheat = 20 deg
  Suction line pressure drop = 2 psi

TABLE 2—OPEN COMPRESSOR RATINGS

REFRIGERANT 12 **8-CYLINDER COMPRESSOR**

Suction Temp (F)	CONDENSING TEMPERATURE (F)														
	90			100			105			110			120		
	Cap. (tons)	Power Input (bhp)	Heat Rej (tons)	Cap. (tons)	Power Input (bhp)	Heat Rej (tons)	Cap. (tons)	Power Input (bhp)	Heat Rej (tons)	Cap. (tons)	Power Input (bhp)	Heat Rej (tons)	Cap. (tons)	Power Input (bhp)	Heat Rej (tons)
−40	5.1	15.3	7.9	4.3	14.3	6.7	—	—	—	—	—	—	—	—	—
−30	8.0	20.9	11.7	6.9	20.3	10.5	6.3	19.9	9.8	5.8	19.4	9.1	—	—	—
−20	11.8	26.1	16.8	10.4	26.0	15.3	9.8	25.9	14.5	9.1	25.7	13.8	7.9	25.1	12.3
−10	16.7	30.8	22.9	15.0	31.4	21.2	14.2	31.5	20.3	13.4	31.6	19.4	11.8	31.6	17.8
0	22.7	35.0	30.1	20.7	36.2	28.1	19.6	36.7	27.1	18.6	37.1	26.1	16.6	37.8	24.1
10	29.9	38.6	38.3	27.5	40.5	36.0	26.2	41.3	34.9	25.0	42.1	33.7	22.5	43.5	31.4
20	38.4	41.5	47.5	35.5	44.1	44.9	34.0	45.3	43.6	32.5	46.5	42.3	29.5	48.7	39.6
30	48.2	43.5	57.7	44.8	47.0	54.7	43.0	48.6	53.2	41.2	50.2	51.7	37.7	53.3	48.6
40	59.5	44.7	68.8	55.4	49.0	65.5	53.3	51.1	63.7	51.3	53.2	62.0	47.1	57.1	58.5
50	72.2	44.9	80.8	67.5	50.1	77.1	65.1	52.7	75.1	62.7	55.3	73.2	57.9	60.2	69.1

NOTES:

1 Where values are not shown for capacity and power input, the operating conditions are beyond the operating limits of the compressor.

2. Capacities are based on liquid subcooling of 15 degrees in the system.

3. Although interpolation is permitted, do not extrapolate. Operation outside limits of the table is not allowed.

4. The refrigerant temperatures shown are the saturation temperatures corresponding to the pressures indicated at the compressor. The actual gas temperatures are higher because of superheat.

5. Ratings are based on operation at 1750 rpm. See table below for multiplying factors for other speeds.

Multiplying Factors for Other Speeds

Rpm	1450	1160
Capacity	0.835	0.674
Bhp	0.798	0.602

RATING BASIS AND CAPACITY MULTIPLIERS FOR REFRIGERANT 12 AND REFRIGERANT 500

Saturated Suction Temp (F)	Rated Suction Gas Temp (F)	CAPACITY MULTIPLIERS									
		Actual Suction Gas Temperature to Compressor (F)									
		−20	−10	0	10	20	30	40	50	60	65
−40	35	0.927	0.940	0.953	0.967	0.980	0.993				
−30	45	0.922	0.934	0.946	0.958	0.970	0.982	0.994			
−20	55	0.920	0.931	0.941	0.952	0.963	0.973	0.984	0.995		
−10	65		0.930	0.939	0.949	0.958	0.967	0.977	0.986	0.995	1.000
0	65			0.940	0.949	0.958	0.968	0.977	0.986	0.995	1.000
10	65				0.950	0.959	0.968	0.977	0.986	0.995	1.000
20	65					0.960	0.969	0.978	0.987	0.996	1.000
30	65						0.970	0.979	0.987	0.996	1.000
40	65							0.987	0.992	0.997	1.000
50	65								0.997	0.999	1.000

Example 1 (contd)

Condensing temperature	= 105 F
Compressor speed	= 1750 rpm
Subcooling (water-cooled condenser), assume	= 5 deg
Refrigerant 12	

Find:

Compressor size, horsepower, heat rejection

Solution:

Suction line loss = approximately 1 4 deg per psi at 30 F

Suction temperature = 30 − (1.4 × 2) = 30 − 2.8
= 27 2 F or 27 F

Compressor capacity correction factors

Superheat correction, 27 F suction, 20 deg superheat
= .985

Subcooling correction = 1 − 005 (15 − 5) = 1 − .05 = 95

$$\text{Equivalent capacity} = \frac{36}{.985 \times 95} = 38\ 5 \text{ tons}$$

Select an 8 cylinder compressor. By interpolation from *Table 2*, at 27 F suction and 105 F condensing,

Compressor capacity	= 40.3 tons
Power input	= 47.6 bhp
Heat rejection	= 50 3 tons

CONDENSING UNIT

There are two types of condensing units, water-cooled and air-cooled.

The selection factors stated under *Compressor Unit* also apply to condensing units with the addition of the following.

For *Water-Cooled Condensing Units:*

1. Condenser Water Source — Water for condensing may be available from such sources as city water, well water, river water, sea water, cooling tower, or spray ponds. If a choice is available, the selection of a water source is generally a matter of economics. The cost of city water is a factor, and also there may be sewage charges for disposing of water used in a once-thru system. The cost of the tower or spray pond as well as the water conditioning cost also influences the choice of a source.
2. Fouling Factor — Fouling factors constitute the thermal resistance to heat flow introduced by scale and other water impurities. Normally, manufacturers rate a water-cooled condenser for various values of water side fouling. Nothing less than a .0005 factor should be used when selecting a condenser, even when good quality water is available, because some surface fouling is present from the beginning of operation. Fouling factors have only a small influence on the capacity of reciprocating compressor equipment as compared with other types of refrigeration equipment. An increase in a scale factor of .0005 reduces the capacity of a condensing unit only about 2%. Ranges of fouling factors used for equipment selection are shown in *Part 5, Water Conditioning.*
3. Entering Condenser Water Temperature — If city water or well water is used for the condensing media in a once-thru system, the maximum water temperature prevailing at the time of maximum refrigeration load is used for the selection. This temperature must be obtained locally from the water company or other local sources. If a cooling tower is used to cool the water, the temperature is based on the design wet-bulb temperature and on the approach which is the difference between the wet-bulb temperature and the water temperature leaving the tower. For further data on tower selection and temperature levels, refer to *Chapter 5.*
4. Water Quantity — The required water quantity may be found from the condenser ratings or may be given as an available quantity. It may be limited by the suggested maximum water tube velocities for various total operating hours per year (*Table 3*). City water quantities generally run from 1-2 gpm per t
   Cooling tower water quantities are usually selected for 3 gpm per ton.

TABLE 3—MAXIMUM SUGGESTED WATER VELOCITIES THRU COOLERS AND CONDENSERS

Normal Operation (hr)	Water Velocity (fps)
1500	15
2000	14
3000	13
4000	12
6000	10
8000	8

A typical rating of a hermetic water-cooled condensing unit is shown in *Table 4.*

For *Air-Cooled Condensing Units:*

1. Entering Air Temperature — The normal summer outdoor design dry-bulb temperature is used as the temperature of the air entering the condenser.
2. Air Flow — The unit must be located so that the flow of air to and from the condenser coil is not impeded. There must be enough space surrounding the unit to prevent recirculation of air. Units with direct-driven propeller fans should not use ductwork for the condenser air because the capacity is lowered and the condensing temperature is raised. Ductwork can be used on some units which use belt-dr i centrifugal condenser fans.

When selecting condensing units, consideration should be given to the variation expected in the load and the types and steps of capacity control available on the unit.

Example 2 — Selection of a Condensing Unit

Given:

Refrigeration load	= 10.0 tons
Saturated suction temperature	= 40 F
Entering condenser water temperature	= 75 F
Fouling factor	= .0005
Refrigerant 12	

Find:

Condensing unit size
Condensing temperature
Power input
Condenser water quantity
Circuiting

Solution:

There are two possible selections depending on the number of condenser passes.

TABLE 4—WATER-COOLED CONDENSING UNIT RATINGS

CAPACITY, POWER INPUT, HEAT REJECTION, CONDENSER WATER **REFRIGERANT 12, COMP MODEL 40, COND MODEL 40**

Cond Temp (F)		Passes	Ent Cond Water Temp (F)	Suction Temperature (F)				
				50	40	30	20	10
				Suction Pressure (psig)				
				46.7	37.0	28.5	21.1	14.6
90	Capacity (tons)			14.4	11.6	9.2	7.2	5.6
	Power Input (kw)			8.1	8.0	7.7	7.3	6.7
	Heat Rejection (tons)			16.7	13.9	11.4	9.3	7.5
	Gpm	8	50	12	10			
		8	60	18	14	11		
		4	60	22	17			
		4	65	30	22			
		4	70	46	32	23	17	
100	Capacity (tons)			13.4	10.8	8.6	6.7	
	Power Input (kw)			9.0	8.7	8.3	7.7	
	Heat Rejection (tons)			16.0	13.3	11.0	8.9	
	Gpm	8	60	11	9			
		8	70	17	13	10		
		4	70	21				
		4	75	28	21			
		4	80	42	30	22		
110	Capacity (tons)			12.4	10.0	8.0	6.2	
	Power Input (kw)			10.0	9.5	8.9	8.2	
	Heat Rejection (tons)			15.3	12.8	10.5	8.6	
	Gpm	8	70	11	9			
		8	80	16	12	10		
		4	80	20				
		4	85	26	20			
		4	90	38	28	21		

NOTES:

1. Where values are not shown for capacity and power input (kw), the operating conditions are beyond operating limits of the compressor.
2. Where values for gpm are not shown, the operating conditions require a water quantity outside the condenser operating limits.
3. The condenser water quantities shown are based on a .0005 fouling factor.
4. Although interpolation is permitted, do not extrapolate. Operation outside the compressor limits of the table is not allowed.
5. For 50-cycle operation, multiply the capacity and power input by 0.83. The condenser water quantities are based on 60-cycle operation. For 50-cycle operation, use the condenser ratings.
6. Condenser leaving water temperature

$$= \text{entering water temp} + \frac{\text{heat rejection (tons)} \times 24}{\text{gpm}}$$

where:

heat rejection (tons) = unit heat rejection (rating tables)

24 = conversion constant

gpm = gal/min (rating table)

7. The refrigerant temperatures shown are the saturation temperatures corresponding to the pressure indicated at the compressor. The actual gas temperature is higher because of superheat.

Example 2 (contd)

	Selection 1	Selection 2
Condensing unit size	40	40
Condensing temperature	100 F	110 F
Power input	8.7 kw	9.5 kw
Condenser water quantity	21 gpm	10.5 gpm
Circuiting	4-pass	8-pass

Four-pass selection is usually used with cooling tower water and eight-pass selection with city water.

WATER-CHILLING UNIT

The factors influencing the selection of a water chilling unit are:

1. *Capacity, Chilled Water Quantity, Temperature Range* – These are related to each other and, when any two are known, the third can be found by the formula:

$$\text{Capacity (tons)} = \frac{\text{gpm} \times \text{temperature range}}{24}$$

Temperature range is the difference between the water temperature entering and leaving the chiller. Capacity is the total load of the chiller, and the water quantity is the design flow; these are generally determined from the dehumidifier(s) selection.

2. *Water Temperature Levels* – The leaving chilled water temperature is usually selected as the entering water temperature required at the load source. The proper determination of the water temperature required for chilled water coils and spray washers is discussed in *Part 6*.

   The entering condenser water temperature is determined by the source of the water, i.e. city water used in a once-thru system, or cooling tower water used in a recirculating system.
3. *Fouling Factors* – The same discussion included under water-cooled condensing units applies to the condenser of a water-chilling unit.

   The fouling factor used for the chiller selection depends on the water conditioning and the system to which the chiller is applied, either an open or closed recirculating system. Water chillers applied to open recirculating chilled water systems should be selected with a minimum fouling factor of .001 in the cooler. For closed recirculating systems, a minimum fouling factor of .0005 should be used. Refer to *Part 5* for recommended fouling factors for different applications and also for information on water conditioning.

A typical rating of a water-cooled water-chilling unit is shown in *Table 5*.

Example 3 – Water-Chilling Unit Selection

Given:

Chilled water quantity = 200 gpm
Leaving chilled water temperature = 44 F
Chilled water temperature rise = 10 degrees
Entering condenser water temperature = 85 F
Fouling factor (cooler and condenser) = .0005

Find:

Unit selection
Condensing temperature
Power input
Condenser water flow rate
Pressure drops thru cooler and condenser

Solution:

$$\text{Load} = \frac{200 \times 10}{24} = 83.3 \text{ tons}$$

Select Model 100. By interpolation:

Condensing temperature = 107.4 F
Power input = 82.0 kw
Condenser load = 106.5 tons

$$Q_c = \frac{106.5}{107.4 - 85} = 4.75$$

For a 3 pass condenser, gpm = 170
Condenser pressure drop = 8 psi
Cooler pressure drop = 10.5 psi

DRIVE SELECTION

Drive selection involves a consideration of types, sizes, starting torque, overload, and starting requirements.

DRIVE TYPE

For an open compressor almost any type of driver can be used. The most common is the polyphase squirrel cage induction motor. Other types used in special cases are d-c motors, wound rotor induction motors, or single phase motors. There may be a need to apply 2-phase motors or 25 or 50 cycle motors for compressors in regions where these electrical conditions exist.

Another type of driver that can be applied to an open compressor is an internal combustion engine, either diesel or natural gas driven. These may be used when an economic analysis of the relative costs of fuel and electrical energy, maintenance and the relative investment required is made. Engines are available commercially in sizes from 10 horsepower and up; they usually are direct-connected.

Hermetic compressor equipment is normally supplied with a squirrel cage induction motor. It may be wound for part-winding starting to provide a method of reducing current inrush; it may be furnished in some sizes with a single phase or two-phase motor, and may be available for 50-cycle duty.

SIZE

Driver selection based on the brake horsepower required at the design operating condition is usually satisfactory for an open compressor application for air conditioning duty. For selections at low design suction temperatures, it is usually necessary to consider the initial pulldown operating condition. The compressor operates at a higher suction temperature at start-up, requiring a higher input during the pulldown. This consideration frequently dictates the size of the motor required rather than the brake horsepower required at the design operating condition.

The motor for a hermetic compressor is selected by the manufacturer to prevent overload when the compressor is running at its normal operating condition.

STARTING TORQUE

It should be noted in the selection of a drive that the required starting torque available by the driver must equal the compressor starting torque only when the compressor is selected to operate at the driver speed (direct-driven). If the design compressor speed is less than the driver speed (as on

TABLE 5—WATER CHILLING UNIT RATINGS

MODEL 100, 60 CYCLE — **ARI Standard Ratings, 85.8 tons, 78.8 kw, 257 gpm condenser**

Leaving Chilled Water Temperature (F) Fouling Factor: .0005 in Cooler Cooling Range: 5 F to 15 F		WATER-COOLED MODEL based on 5 F subcooling					CONDENSERLESS MODEL based on 15 F subcooling				
		Compressor Saturated Discharge Temperature (F)					Compressor Saturated Discharge Temperature (F)				
		90	95	100	105	110	105	110	120	130	135
40	Capacity (tons)	85.3	82.8	80.3	78.1	75.8	81.6	79.1	73.6	67.6	64.4
	Power Input (kw)	71.2	74.0	76.4	79.0	81.9	78.9	81.5	86.6	93.8	97.4
	Condenser Load (tons)	105.5	103.8	102.0	100.6	99.1	104.1	102.3	98.3	94.3	92.2
42	Capacity (tons)	88.6	86.1	83.6	81.4	79.0	84.6	82.2	76.5	70.3	66.9
	Power Input (kw)	71.5	74.2	77.1	79.6	82.5	79.5	82.4	87.6	95.2	99.0
	Condenser Load (tons)	108.9	107.2	105.5	104.0	102.5	107.2	105.6	101.4	97.4	95.1
44	Capacity (tons)	92.0	89.4	87.0	84.6	81.9	87.9	85.3	79.6	73.1	69.6
	Power Input (kw)	71.8	74.7	77.5	80.4	83.4	80.2	83.2	88.6	97.0	100.9
	Condenser Load (tons)	112.4	110.6	109.0	107.5	105.6	110.7	109.0	104.8	100.7	98.3
46	Capacity (tons)	95.5	92.6	90.3	87.9	85.0	91.3	88.4	82.5	76.0	72.4
	Power Input (kw)	72.1	75.1	78.1	81.0	84.3	80.9	84.1	89.7	98.4	102.5
	Condenser Load (tons)	116.0	114.0	112.5	110.9	109.0	114.2	112.3	108.0	104.0	101.6
48	Capacity (tons)	98.8	96.1	93.5	90.9	88.1	94.5	91.6	85.6	78.9	75.1
	Power Input (kw)	72.1	75.3	78.4	81.6	84.9	81.4	84.6	90.7	99.7	104.0
	Condenser Load (tons)	119.3	117.5	115.8	114.1	112.3	117.7	115.7	111.4	107.3	104.7
50	Capacity (tons)	102.1	99.4	97.1	94.1	91.3	97.6	94.7	88.8	81.9	78.0
	Power Input (kw)	72.2	75.5	78.8	82.1	85.5	81.9	85.3	91.5	100.9	105.6
	Condenser Load (tons)	122.6	120.9	119.5	117.5	115.6	120.8	118.9	114.8	110.6	108.1

ARI STANDARD RATINGS

ARI Standard 590 for Reciprocating Liquid Chilling Packages requires that Standard Ratings be published for established operating conditions

The ARI rating shown above the rating table applies to water-cooled models only, and is based on 54 F to 44 F chilled water and 85 F to 95 F condenser water with a .0005 fouling factor in both exchangers

ARI ratings for the condenserless models are indicated in the rating table by heavy lined boxes and are based on a .0005 fouling factor in the cooler

COOLING FOULING FACTOR ADJUSTMENTS

Fouling Factor	Capacity	Power Input (kw)
Clean	1.02	.98
.0005	1.00	1.00

Condenser fouling factors are covered in the curves below.

RATINGS IN THE TABLES are based on the following:

1. Cooling range: 5 to 15 degrees
2. Fouling factor: 0005 in the cooler
3. Liquid subcooling: water-cooled models, 5 degrees in condenser
   condenserless models, 15 degrees in condenser

(If a condenser is selected for less than 15 degrees of subcooling, adjust as follows: Multiply tabular capacity rating by 0.94. Then adjust this result upward by 0.4 percent for each one degree of available subcooling.)

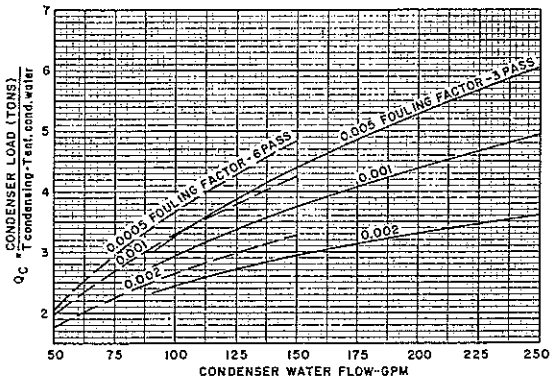
FORMULAS

$$\text{Capacity (tons)} = \frac{24}{\text{gpm} \times \text{temp drop}}$$

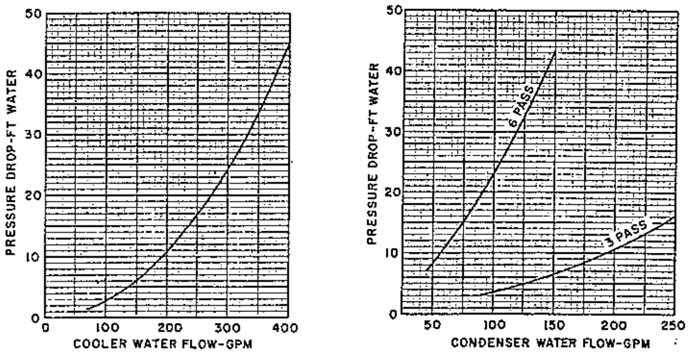
$$Q_c = \frac{\text{condenser load (tons)}}{\text{condensing temp} - \text{entering condenser water temp}}$$

$$\text{Condenser water rise} = \frac{24 \times \text{condenser load}}{\text{gpm}}$$

CONDENSER GPM



PRESSURE DROP



belt-drive units), the starting torque requirements are reduced in proportion to the speed ratio between the compressor and drive, because of the mechanical advantage available to the driver.

The following formulas are useful in analyzing motor performance with respect to starting torque requirements:

1. Motor full load torque (lb ft) $= \dfrac{5250 \times \text{hp}}{\text{motor rpm}}$

2. Starting torque available at motor (lb ft)

$$= \frac{5250 \times \text{motor hp} \times \text{percent starting torque} \times 0.81}{\text{motor rpm} \times 100}$$

3. Starting torque available at compressor (lb ft)

$$= \frac{5250 \times \text{motor hp} \times \text{percent starting torque} \times 0.81}{\text{compressor rpm} \times 100}$$

4. Minimum motor horsepower required to start compressor

$$= \frac{\text{comp starting torque (lb ft)} \times \text{comp rpm} \times 100}{5250 \times \text{percent starting torque} \times 0.81}$$

(Incorporation of the 0.81 factor in formulas 2, 3 and 4 allows up to 10% drop in voltage at the motor terminals during the starting periods.)

The starting torque which a motor develops is proportional to the square of the voltage at the motor terminals. Thus, at half voltage a motor develops only one-quarter of the torque developed at full voltage. To make starting possible, the voltage at the motor terminals must be high enough to provide the required starting torque.

A compressor equipped with a capacity control may be supplied with a normal starting torque motor (NEMA design B) when it is started partially loaded. A compressor not equipped with capacity control should be provided with a high starting torque motor (NEMA design C). The compressor manufacturer should be consulted as to the starting torque requirements of the compressor.

With reduced voltage starting, it is usually necessary to use a high starting torque motor when the starting inrush must be kept to the lowest possible value coincident with actual starting of the compressor. *Table 6* shows NEMA starting torque and locked rotor current values for both normal and high torque motors in the 5-200 horsepower range. The full voltage starting torque (locked rotor torque) is usually expressed in percent of the torque delivered by the motor at full load, full speed and at rated voltage and frequency. For convenience, the actual torque (pound feet) of the motor is also shown in the table.

OVERLOAD

NEMA standards permit continuous overloading of 40 C rise open squirrel cage motors up to 15% over nameplate rating when operated at full rated voltage, frequency and ambient temperatures not exceeding 40 C. Whether or not part of this 15% service factor should be used in making an initial motor selection depends on having available accurate information concerning local voltage and frequency variation, ambient temperature, compressor speed, and maximum suction and condensing pressures at which the compressor operates. Where conditions of voltage, frequency, and ambient temperatures are comparatively unknown, *motors should positively not be selected overloaded.* Selection [illegible] operation above full load current may shorten the life of a motor as much as 50%.

BELT DRIVE

When using a belt drive for a compressor, it is recommended that 3% be added to the direct drive brake horsepower ratings to obtain the approximate motor brake horsepower required.

For more complete information on motors and their characteristics, refer to *Part 8*. For information on other types of drives, refer to the manufacturer.

STARTING EQUIPMENT

HERMETIC COMPRESSOR

Normally, hermetic compressor equipment is completely wired at the factory and supplied with either a standard motor and across-the-line starter or a part-winding motor and increment type starter, designed specifically for use with the compressor. The increment type starter can be augmented with a resistance type accessory to make it a three-step starter with a lower inrush current than with a part-winding type alone.

OPEN COMPRESSOR

Whenever possible, across-the-line starting is most desirable because it is less costly and less troublesome than the complicated reduced voltage equipment. However, limitations of the power distribution system capacity often require the use of a reduced voltage starter for compressor motors above a certain size. In each case the power company concerned should be consulted and a rating obtained with respect to the particular application.

When reduced voltage starting must be used, increment starting is the least expensive because it requires no voltage-reducing elements such as transformers or resistors. Although the most expensive

TABLE 6—NEMA STARTING TORQUE VALUES

STANDARD OPEN SQUIRREL CAGE MOTORS, 60 CYCLE, 3 PHASE

MOTOR HP	FULL LOAD RPM	MINIMUM STARTING TORQUE (At Full Voltage)				LOCKED ROTOR CURRENT AT 220 VOLTS (amps)
		Design "B" Normal Starting Torque Normal Starting Current		Design "C" High Starting Torque Normal Starting Current		
		Percent of Full Load	Pound Feet	Percent of Full Load	Pound Feet	
5	1750	185	27.8	250	37.4	90
7½	1750	175	39.4	250	56.2	120
10	1750	175	52.5	250	74	150
15	1750	165	74.2	225	101	220
20	1750	150	90	200	120	290
25	1750	150	112	200	150	365
30	1750	150	135	200	180	435
40	1750	150	180	200	240	580
50	1750	150	225	200	300	725
60	1750	150	270	200	360	870
75	1750	150	337	200	450	1085
	1160	135	458	200	680	1085
	870	125	565	200	906	1085
100	1750	125	375	200	600	1450
	1160	125	565	200	906	1450
	870	125	755	200	1205	1450
125	1750	110	410	200	750	1815
	1160	125	705	200	1130	1815
	870	125	940	200	1510	1815
150	1750	110	495	200	900	2170
	1160	125	850	200	1355	2170
	870	125	1130	200	1810	2170
200	1750	100	600	200	1200	2900
	1160	125	1130	200	1810	2900
	870	125	1510	200	2410	2900

type, auto transformer starters obtain the most effective reduction of inrush current drawn from the line. Primary resistance starters are less expensive and obtain a smaller degree of reduction of inrush current. Refer to *Part 8* for data on starters.

CONTROLS

Compressor control consists mainly of capacity control, safety controls, and the method of compressor operation.

COMPRESSOR CAPACITY CONTROL

Various methods of capacity control are available from different compressor manufacturers. A brief description and some of the advantages or disadvantages of these methods are included here.

Suction-Valve-Lift Unloading

This unloading is accomplished by unseating the suction valves of certain cylinders in the compressor so that compression cannot take place. This is inherently the most efficient method of capacity control since passage of the refrigerant vapor in and out of the cylinder thru the suction valves without compression involves smaller losses than other methods. The compressors are generally controlled in steps down to one fourth or one third of full load capacity, depending on the number of cylinders. The cylinders may be unloaded internally under the control of suction pressure or externally from a thermostat or pressurestat. *Figure 5* illustrates a method of this type of capacity control.

Cylinder Head Bypass

On multi-cylinder compressors one or more cylinders or banks of cylinders may be made ineffective by a bypass of gas from the cylinder discharge to the intake port (*Fig. 6*). A check valve is installed to separate the inactive cylinders from active ones. A solenoid valve can be installed to operate the by-

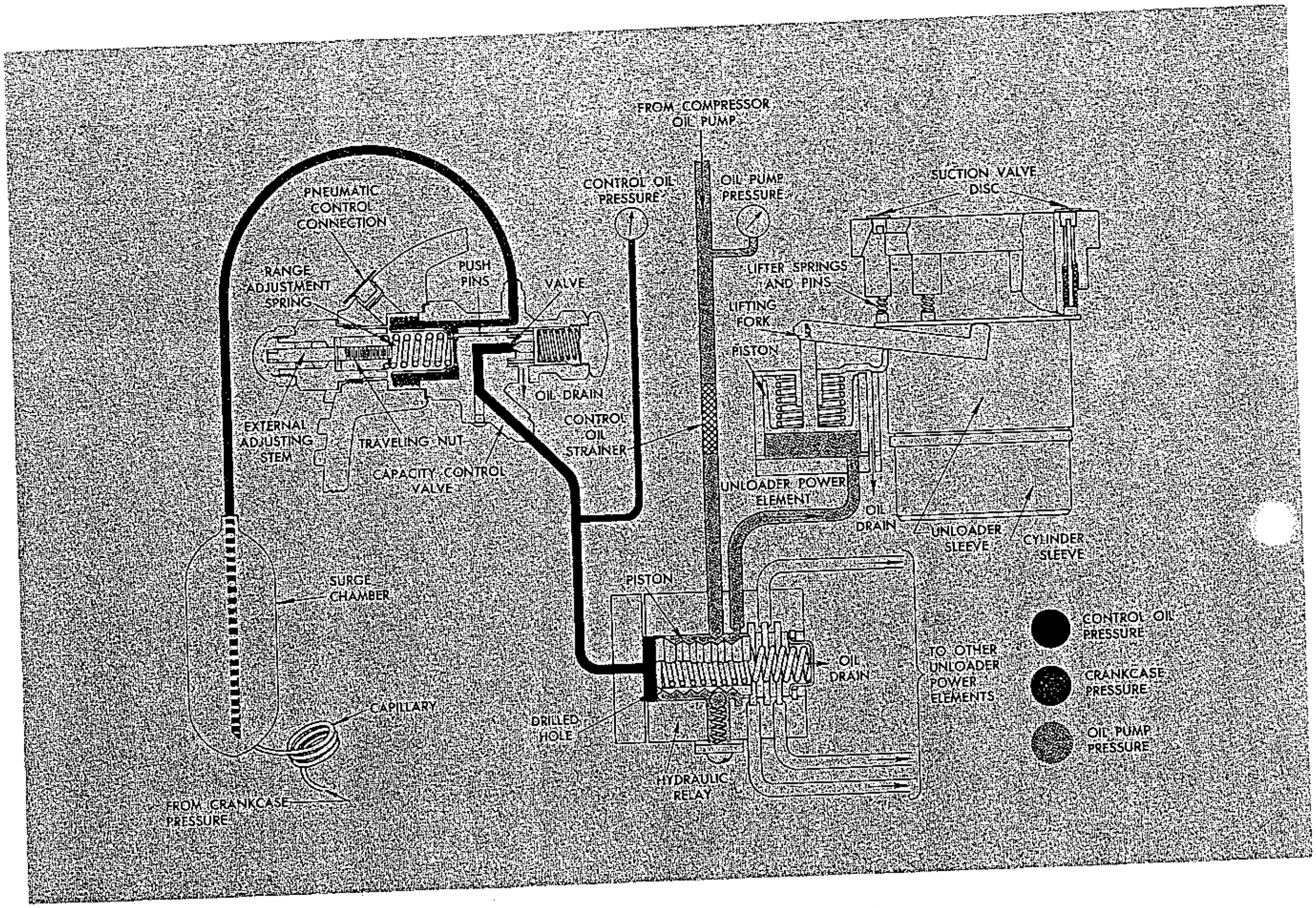
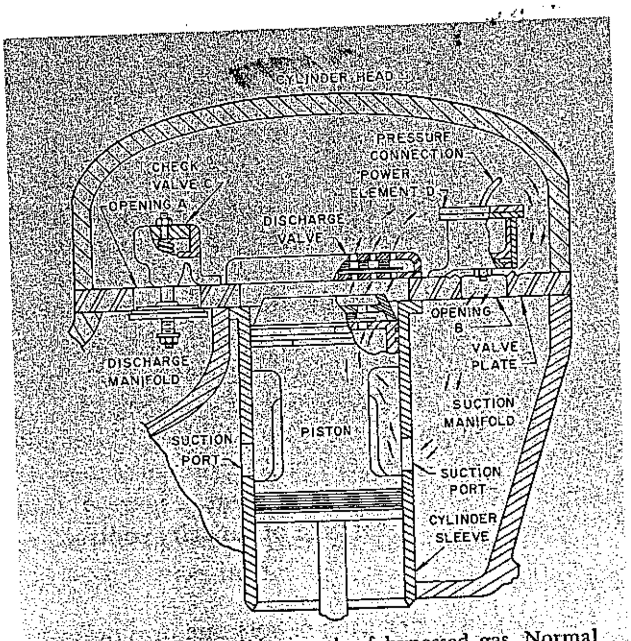


FIG. 5 — SUCTION-VALVE-LIFT UNLOADING



NOTE: Arrows indicate path of bypassed gas. Normal gas passage is thru check valve C to discharge manifold.

FIG. 6 — CYLINDER HEAD BYPASS

pass, thus permitting automatic control by a thermostat or pressurestat. Gas passes thru the inactive cylinders but is not compressed; therefore, the only loss is thru the valves, cylinders, and connections. Because of these losses, the power requirement does not decrease proportionally with the capacity.

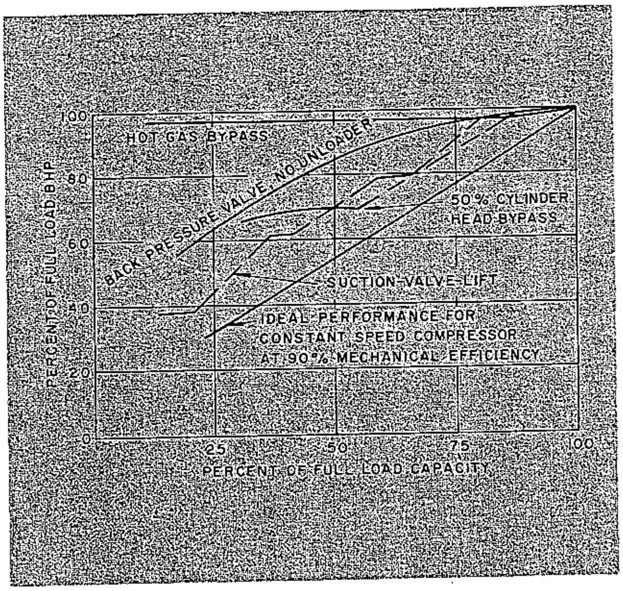
Speed Control

Compressor capacity is almost directly proportional to the speed while the compressor brake horsepower is proportional to the ratio of the speeds raised to a power of 1.0 to 1.3, depending on the compressor design. The speed control can be obtained by using a multi-speed motor which provides two or three speeds, or by using an internal combustion engine which can produce multiple speeds. Care must be taken that the compressor is not operated at a speed below the range for proper operation of the lubricating system.

Multiple Units

The use of multiple compressors for capacity control (*Fig. 7*) has the following advantages: (1) singl speed motors may be selected and operated con

CHART 1—POWER-SAVING CHARACTERISTICS OF TYPICAL COMPRESSOR LOADING AND UNLOADING DEVICES



creased so that the compressor unloads at a higher than normal suction temperature.

SAFETY CONTROLS

There are several safety controls which can be applied to reciprocating compressors. A brief explanation of the function of each is included.

Oil Safety Switch

This switch (*Fig. 9*) may be used on compressors with pressure type lubrication It stops the compressor if there is a lubrication failure due to either oil leaks from the system, a clogged strainer stopping the oil intake to the pump, excessive refrigerant in the crankcase, or insufficient oil pressure. Provision must be made to bypass this switch on start-up or to use a time delay switch when the oil pump is direct-driven from the compressor shaft.

Low Pressure Switch

This switch is used to stop the compressor when the suction pressure is reduced to a point which could produce freezing in a water chiller, or which would permit operation beyond the prescribed operating limits of the compressor.

High Pressure Switch

This switch is used to stop the compressor when the discharge pressure rises above the prescribed limits because of either inadequate condensing, an overcharge of refrigerant air in the system, or any other reason. It is usually combined with the low pressure switch into one control called a dual-pressure switch (*Fig. 10*).

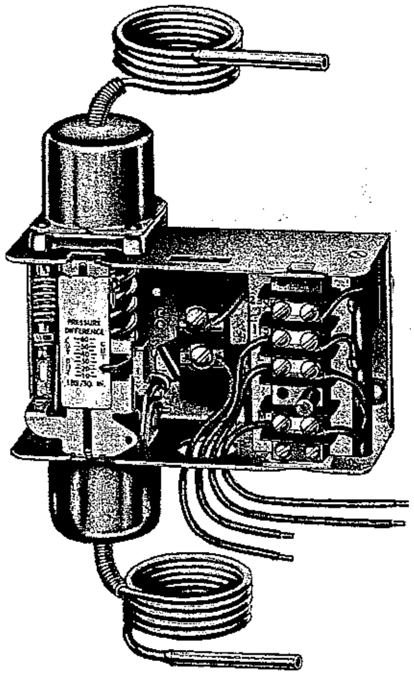


FIG. 9 — OIL SAFETY SWITCH

Chilled Water Safety Switch

This switch is used with water-chilling units to stop the compressor when the water temperature in the chiller approaches the freezing point.

Time Delay Relay

This relay should be used on hermetic compressors, and can be used on open compressors to prevent short cycling of the unit. The relay should be set to prevent the starting of the compressor until an elapse of time (such as 5 minutes) after the compressor has been shut down, due to action of one of the safety or operating controls. This reduces the chances of overheating the motor, and eliminates a possible burnout because of frequency of starting.

Motor Temperature Switch

This switch is used on hermetic compressors to stop the compressor when the temperature in the motor windings becomes excessive

Motor Overloads

These overloads are included in the wiring of the compressor circuit to stop the compressor when the motor draws excessive current.

COMPRESSOR OPERATION

Reciprocating compressor operation must prevent excessive accumulation of liquid refrigerant in the crankcase during off cycles. This minimizes the rapid evaporation of the refrigerant on start-up, resulting in foaming of the oil and loss of lubrication.

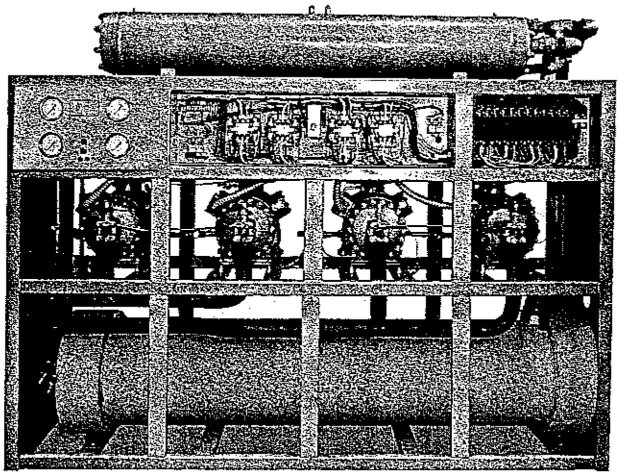


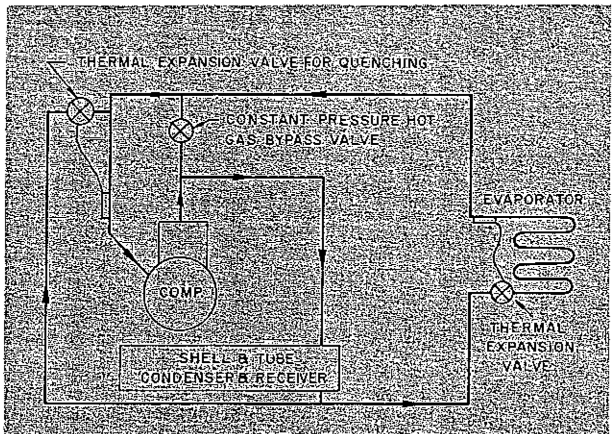
FIG. 7 — MULTIPLE COMPRESSORS

tinuously at their best efficiency; (2) stand-by equipment is available which allows partial load operation if one of the machines breaks down; (3) compressors may be started in sequence to limit the current inrush if time-delay devices are employed. The compressors may or may not be interconnected as conditions require. Thermostats or pressurestats may be used to start and stop compressors in accordance with load demands.

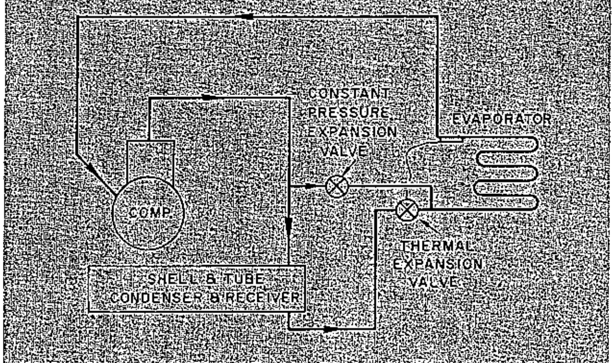
Hot Gas Bypass

Another method of capacity control is to load the compressor artificially. This can be accomplished by transferring heat to the suction gas in the form of a hot gas bypass. The discharge is connected to the low side thru a constant pressure valve which admits hot gas to the low side as the evaporator pressure tends to drop, thus maintaining a constant suction pressure. *Figure 8* shows three arrangements of hot gas bypasses. Since this is a method of loading and not unloading the compressor, the compressor brake horsepower remains fairly constant.

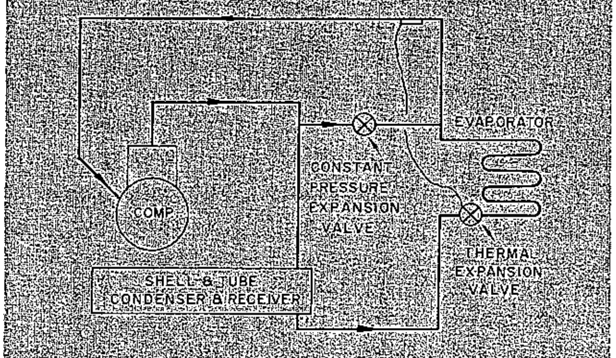
Though not a method of capacity control, a back pressure valve does permit a constant speed compressor to operate at lower capacities and horsepowers while maintaining a constant evaporator temperature. As the load is decreased, the evaporator temperature is reduced. This causes the back pressure valve to start to close, creating a restriction between the evaporator and the compressor suction, which causes a reduction in suction pressure while the evaporator pressure remains close to design. As the suction pressure decreases, the density of the suction gas decreases, and the weight flow of refrigerant is reduced, thus lowering the capacity of the compressor.


THERMAL EXPANSION VALVE FOR QUENCHING
CONSTANT PRESSURE HOT GAS BYPASS VALVE
EVAPORATOR
COMP
THERMAL EXPANSION VALVE
SHELL & TUBE CONDENSER & RECEIVER

HOT GAS BYPASS WITH LIQUID QUENCH


CONSTANT PRESSURE EXPANSION VALVE
EVAPORATOR
COMP
THERMAL EXPANSION VALVE
SHELL & TUBE CONDENSER & RECEIVER

HOT GAS BYPASS TO ENTRANCE OF EVAPORATOR

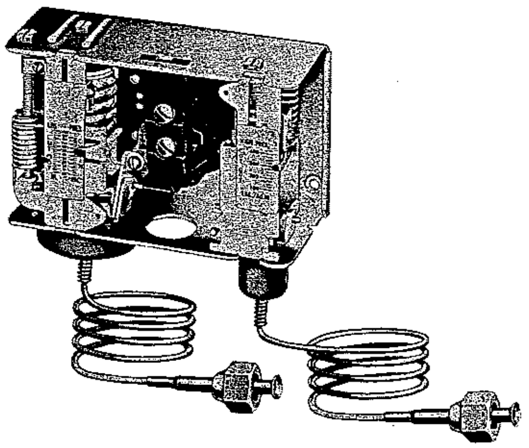

EVAPORATOR
CONSTANT PRESSURE EXPANSION VALVE
COMP
THERMAL EXPANSION VALVE
SHELL & TUBE CONDENSER & RECEIVER

HOT GAS BYPASS TO EXIT OF EVAPORATOR

FIG. 8 — HOT GAS BYPASSES

Chart 1 shows the power-saving characteristics of typical compressor loading and unloading devices.

Compensation of the compressor capacity from a room thermostat or a chilled water thermostat is used when control of the temperature must be closer than the 8-10 degree swing normally obtained when using only suction-valve-lift unloading. This compensation is accomplished by resetting the control point at which the compressor unloads. For example, as the temperature at the room thermostat or chilled water thermostat decreases, the control point is in-



Courtesy of Penn Controls, Inc.

Fig. 10 — Dual Pressure Switch

Methods of Control

Either of the following methods of control may be used to prevent this accumulation of excess refrigerant:

1. *Automatic Pumpdown Control (DX Systems)*— The most effective and most common means of keeping liquid out of the crankcase during system shutdown periods is to operate the compressor on automatic pumpdown control. It is most practical on small systems using a single DX evaporator. A typical wiring diagram for this control is shown in *Fig. 11*. The recommended control arrangement involves the following devices and provisions:
   - a. A tight-closing solenoid valve in the main liquid line or in the branch to each evaporator.
   - b. Compressor operation thru a low pressure switch providing for pumpdown whenever the valve closes, whether the balance of the system is in operation or not.
   - c. Electrical interlock of the liquid solenoid valve(s) with the evaporator fan or water chiller pump, so that the refrigerant flow is stopped when either the fan or the pump is out of operation.
   - d. Electrical interlock of the refrigerant solenoid valve(s) with the safety devices (high pressure cutout, oil safety switch and motor overloads) so that the valve(s) closes when the compressor stops, due to any one of these safety devices.
   - e. Low pressurestat settings, such that the cut-in point corresponds to a saturated refrigerant temperature lower than any ambient air temperature to which the compressor may be subjected.
2. *Crankcase Oil Heater With Single Pumpout At The End of Each Operating Cycle (DX Systems)* — This arrangement is not as positive in keeping liquid refrigerant out of the crankcase as automatic pumpdown control, but it is a substitute where pumpdown control (resulting in compressor cycling) meets with customer objections. A typical schematic wiring diagram for single pumpout control with crankcase heater is shown in *Fig. 12*. The use of this method at the end of each operating cycle requires the following:
   - a. A tight-closing solenoid valve in the main liquid line or in the branch to each evaporator.

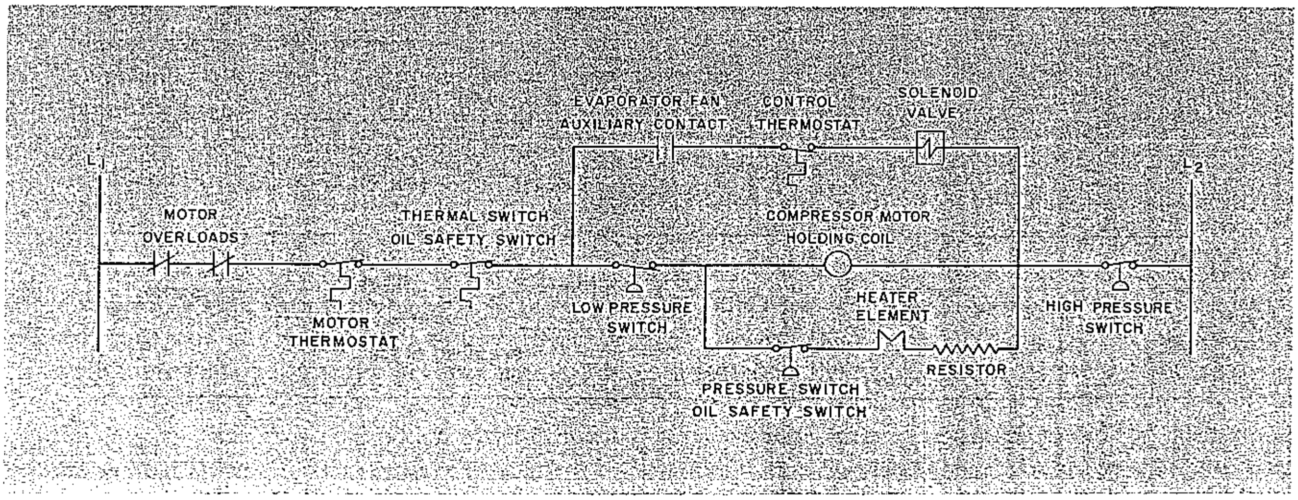


Fig. 11 — Typical Automatic Pumpdown, Wiring Diagram

b. A relay or an auxiliary contact of the compressor motor starter to maintain compressor operation until the low pressure switch opens.

c. A relay or auxiliary contact for energizing the crankcase heater during the compressor off cycle and de-energizing it during the compressor on cycle.

d. Electrical interlock of the refrigerant solenoid valve(s) with the evaporator fan or chilled water pump, so that the refrigerant flow is stopped when either the fan or pump are out of operation.

e. Electrical interlock of the refrigerant solenoid valves(s) with the safety devices, (high pressure cutout, oil safety switch and motor overloads), so that the valve(s) closes when the compressor stops, due to any one of these safety devices.

3. *Compressor Control With Flooded Evaporators* – Neither automatic pumpdown control nor single pumpout operation is practical in systems employing flooded evaporators unless suction line solenoid valves are added to the system. Therefore, with flooded evaporators the following arrangements are often used:

   a. Manual operation (Item 4). No crankcase heaters are required.

   b. Automatic control from temperature controllers or other devices, provided crankcase heaters are used and energized on the off cycles, and the liquid solenoid valve is closed whenever the compressor is stopped. Where water cooling of the compressor head is employed, a solenoid valve in the water supply line closes whenever the compressor is stopped.

   c. Item b, with the added precaution of a single pumpout of the compressor for night or weekend shutdowns. This can be accomplished by manually closing the compressor suction stop valve.

4. *Manual Compressor Operation* – Compressors may be controlled manually without the use of automatic pumpdown control, or by single pumpout and a crankcase heater, provided the system is under the control of a qualified operator at all times. The operator pumps down system by use of the manual valves, and keeps the liquid, suction and discharge valves closed when the machine is not operating.

Effect of a Short Operating Cycle

It is characteristic of the reciprocating compressor operation that oil leaves the crankcase at an accelerated rate immediately after starting. Therefore, each start must be followed by a sufficiently long operating period to permit the regain of the oil level. Operation under control of a room thermostat generally provides enough operating time in most cases. However, if the compressor is controlled in response to a thermostat located in the supply air or in the water leaving a water chiller, a rapid cycle may result. This thermostat should have a differential wide enough so that the running cycle is not less than seven or eight minutes.

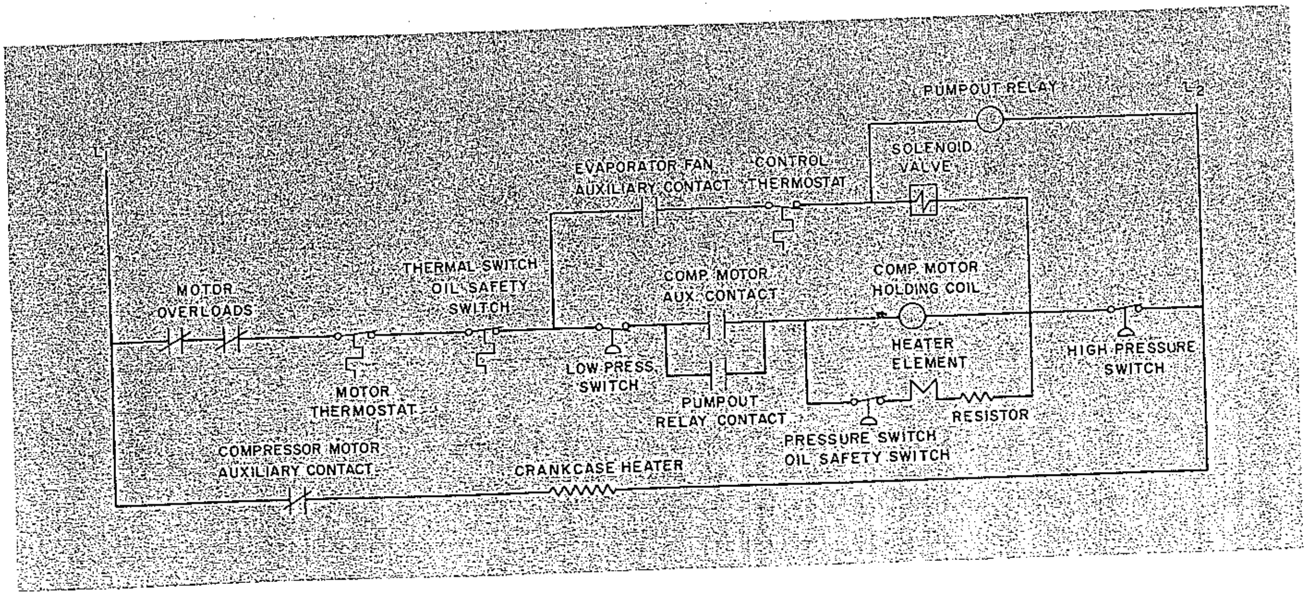


Fig. 12 – Typical Single Pumpout With Crankcase Heater, Wiring Diagram

ACCESSORIES

The following accessories may be required and can be supplied with the compressors.

1. *Coupling* — Used with an open compressor when driven at motor speed.
2. *Belt Drive Package* — Used with an open compressor utilizing a fly wheel and motor pulley for driving the compressor at any specific speed.
3. *Vibration Isolators* — Used to isolate compressor units, condensing units or water chilling units to reduce transmission of noise and vibration to the floor or building structure.
4. *Crankcase Heaters* — Used to keep the oil warm in the crankcase when the compressor is not running. This heating prevents the oil from absorbing refrigerant to an excessive degree, thus maintaining its full lubricating and protective qualities.
5. *Water-Cooled Compressor Heads* — Used to prevent excessive temperatures at the discharge valve. They are usually required when a compression ratio of 5:1 is exceeded when using Refrigerant 22. To prevent condensation of refrigerant in the cylinders, water flow must be stopped when the compressor stops. The maximum leaving water temperature should be 100 F.
6. *Suction Strainer* — Used to prevent foreign particles from entering the compressor. It is important that the system be thoroughly cleaned before start-up.
7. *Structural Steel Base* — Used to mount the compressor unit, condensing unit, or water chilling unit as a completely fabricated assembly, for ease of installation.
8. *Crankcase Connections* — Used to interconnect two or more compressors (connected to the same system) to return oil equally to all compressors
9. *Muffler* — Used to minimize refrigerant noise. It should be installed so oil is not trapped.

INSULATION

Cold surfaces such as the cooler shell of a water chilling unit and the suction pipe should be insulated to prevent dripping where this condition creates a nuisance or causes damage. The thickness should be such that the temperature of the outer surface is slightly higher than the expected dewpoint of the surrounding air. An external vapor barrier should be used to prevent leakage of vapor into the insulation. Cellular plastic or cellular glass type of insulation can be used since they have a high resistance to water and water vapor, and are good insulators.

Hot gas lines are not insulated unless there is some danger of receiving burns by contact with the lines. If this is to be prevented, the hot gas lines may be insulated up to 5 feet from the floor with a high temperature insulation such as magnesia.

Liquid lines should not be insulated unless heat can be picked up from the surrounding air, i.e. where they are installed exposed to the direct sunlight for a considerable distance or installed in boiler rooms. Insulation may also be used at the outlet of a liquid suction interchanger to preserve the subcooling effect.

LOCATION

The location of a reciprocating refrigeration machine should be carefully planned; it directly influences the economic and sound level aspects of any system.

In general, the compressor should be located in a clean, dry, well-ventilated space. Cleanliness and absence of dampness insure long life to motors and belts, and reduce the necessity of frequent painting of exposed piping. If natural ventilation is inadequate or cannot be supplied thru windows and doors, forced ventilation thru ductwork should be provided. It is essential that the starter and open motors have adequate ventilation to prevent overheating of the starter and overloading of the motor.

Space should be available at the end of all replaceable tube coolers and/or condensers so that the tubes can be cleaned or replaced. Adequate space should be left around and over the compressor for servicing; it should be accessible from all sides. Sufficient space should be left above the unit for removing cylinders, and on either side to permit removal of the flywheel and crankshaft.

The unit should be protected so that the water-cooled condenser, water lines and accessories are not subject to freezing during winter shutdown periods

The machine should be located near the equipment it serves to effect a minimum first cost system. However, there may be cases where the machine must be located elsewhere because of space, structural, or sound considerations. The machine should be located where moderate sound levels can be accepted; otherwise special sound proofing may be required when adjacent to low ambient sound level areas such as executive offices or conference rooms.

In new construction the floor framing of the equipment room should be laid out by the architect or consulting engineer to match equipment supports, and should be designed for weights, reactions, and speeds furnished by the equipment supplier. This framing transfers equipment loads to the building columns.

In existing buildings, use of existing floor slabs should be carefully studied. Any deflection in the floor due to weight of the equipment together with vibrations transmitted thru equipment isolation may result in magnifying the vibration in the building structure. Supplementary steel framing to transfer all equipment loads to building columns may be required by the architect or consulting engineer.

LAYOUT

In the layout of a reciprocating refrigeration machine, consideration should be given to foundations and electrical connections.

FOUNDATIONS

Where a foundation is required for a machine, it must be of ample size, have proper proportions, and be constructed of first class materials.

The functions of a foundation are the following:

1. Support and distribute the weight of the machine over a sufficient area so that it is rigidly located.
2. Absorb the forces produced by the rotating and reciprocating parts of the machine. Forces produced by the reciprocating parts act along the center line of the piston. Those produced by rotating parts act radially in all directions from the center of the crank. The magnitude of these forces depends on the weight of the parts and the speed of the machine. These forces produce perceptible and perhaps objectionable vibration of the machine and foundation if the latter does not have sufficient mass.
3. Hold the machine rigid against unbalanced forces produced by the pull of the belts or other sources.

ELECTRICAL CONNECTIONS

Rigid conduit should never be fastened directly to the compressor or base because it can transmit vibration. Instead, flexible conduit should be used.

UNIT ISOLATION

Isolators are of value not only on upper floors in the prevention of transmission of vibration to the building structure, but also on concrete basement floors.

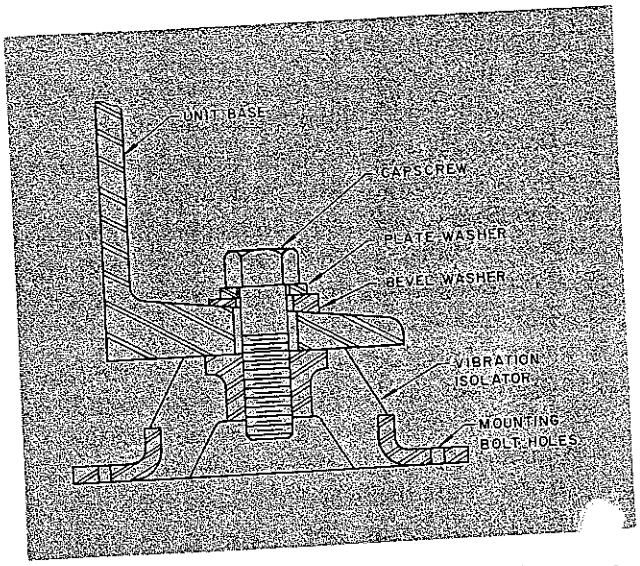


FIG. 13 – TYPICAL VIBRATION ISOLATOR MOUNTING

Two common ways in which vibration is transmitted from reciprocating refrigeration compressors to building structures are the following:

1. Thru the compressor base directly to the building structure.
2. Thru refrigerant and condenser water piping directly to the building structure (*Part 3*).

Figure 13 shows a typical vibration isolator mounting.

When using belt drive compressors, a greater deflection of the isolator is required to maintain the same effectiveness of isolation, as the compressor rpm decreases.

While the standard isolator package is suitable for most applications, for example, on ground or basement floors, a superior isolation may sometime be required particularly for upper floor installation where complete freedom from vibration transmis sion and unusual quietness is a prerequisite.

Where the isolation problem is critical as i upper floor equipment rooms, spring mountings ar recommended. They should be used in conjunctio with a stabilizing mass such as a concrete found tion or steel base, and should be selected for th lowest disturbing frequency, which is the con pressor speed.

The important considerations in selecting su isolating equipment are these:

1. The isolator must permit sufficient deflecti under a load to produce high isolating e ciency.

2. The isolator must retain its resilience; that is, it must not become permanently deformed as otherwise its efficiency would be impaired.
3. The isolator must be structurally suitable to the load imposed and must be applied to a base which distributes the load effectively. Inequality of weight distribution, i.e. a flywheel projecting beyond the base, often necessitates various sizes of isolator units under a common base to produce the required deflection at all points. Manufacturers of isolating equipment publish ratings and physical details of these units. Ratings are published in terms of deflection under load and maximum permissible loading. With an understanding of the problem the designer can use such data to select the necessary equipment for his particular application.

CHAPTER 2. CENTRIFUGAL REFRIGERATION MACHINE

Centrifugal refrigeration equipment is built for heavy duty continuous operation and has a reputation for dependability in all types of commercial and industrial applications.

This chapter presents data to guide the engineer in the practical application and layout of centrifugal refrigeration machines used for cooling water or brine at comfort air conditioning temperature levels.

A centrifugal refrigeration machine consists basically of a centrifugal compressor, a cooler and a condenser. The compressor uses centrifugal force to raise the pressure of a continuous flow of refrigerant gas from the evaporator pressure to the condenser pressure. A centrifugal compressor handles high volumes of gas and, therefore, can use refrigerants having high specific volumes. The cooler is usually a shell-and-tube heat exchanger with the refrigerant in the shell side. The condenser is also a shell-and-tube type utilizing water as a means of condensing; it may be an air-cooled or evaporative condenser for special applications.

TYPES OF CENTRIFUGAL REFRIGERATION MACHINES

Centrifugal refrigeration machines may be classified by the type of compressor:

1. Open compressors have a shaft which projects outside the compressor housing, requiring a seal to isolate the refrigerant space from the atmosphere.
2. Hermetic compressors have the driver built into the unit, completely isolating the refrigerant space from the atmosphere.

OPEN MACHINE

Open type equipment may be obtained for refrigeration duty in single units up to approximately 4500 tons capacity at air conditioning temperature levels. The compressor is normally designed with one or two stages, and is driven by a constant or variable speed drive. Compressors are usually driven at speeds above 3,000 rpm and may operate up to 18,000 rpm.

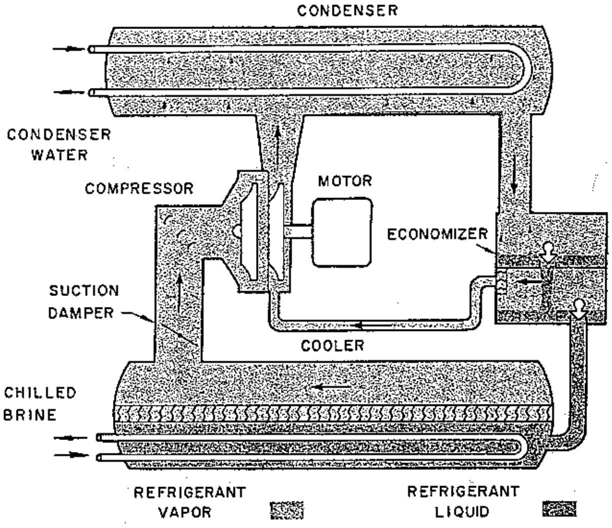


FIG. 14 — OPEN CENTRIFUGAL MACHINE

The centrifugal drive may be an electric motor, steam turbine, gas engine, gas turbine or diesel engine. An electric motor, gas engine or diesel engine usually requires a speed-increasing gear between the drive and the compressor. Gas turbine operating at high speeds may require a speed-decreasing gear between the turbine and the machine. Steam turbines are usually directly connected to the compressor.

Figure 14 illustrates the three basic components, compressor, cooler and condenser, as well as the refrigerant cycle.

Capacity can be varied to match the load by means of a constant speed drive with variable inlet guide vanes or suction damper control, or a variable speed drive with the suction damper control.

HERMETIC MACHINE

Standard hermetic equipment may be obtained in single units up to approximately 2000 tons capacity. They are normally designed with either one or two stages and are driven at a single speed. The drive motor may be either refrigerant- or water-cooled.

A hermetic machine may be driven at motor speed or, by means of a speed-increasing gear between the motor and compressor, at a single higher

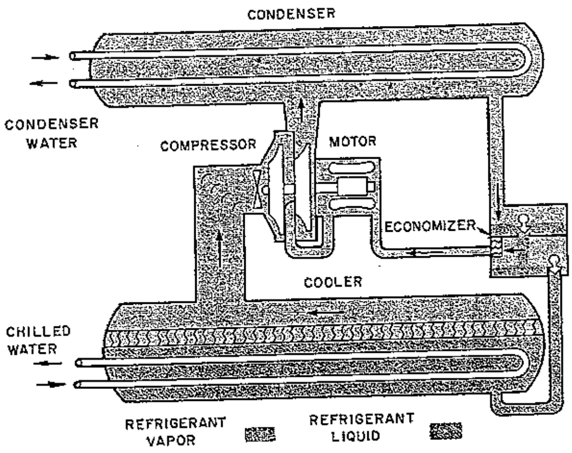


FIG. 15 — HERMETIC CENTRIFUGAL MACHINE

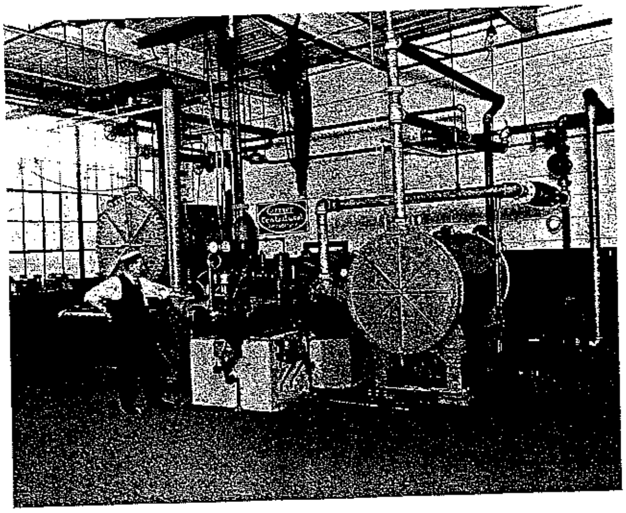


FIG. 16 — THE ORIGINAL CENTRIFUGAL MACHINE (1922)

speed. *Figure 15* illustrates the three basic components, compressor-motor, cooler and condenser, as well as the refrigerant cycle.

Most machines use variable inlet guide vanes for capacity control.

APPLICATION

Centrifugal refrigeration machines were developed to fill the need for single refrigeration units of large capacity. A single centrifugal machine can be used in place of many reciprocating units.

Since the original one was installed (*Fig. 16*), centrifugal refrigeration machines have been known for:

1. Reliability
2. Compactness
3. Low maintenance costs
4. Long life
5. Ease of operation
6. Quietness

Open centrifugal machines are essentially multipurpose machines. They are used in special and industrial applications requiring higher temperature lifts than normally encountered at air conditioning levels. They are flexible in regard to speed selection and staging, and are used for standard water chilling applications where one or more large capacity machines are required, or where a steam turbine, gas engine, gas turbine, diesel engine or special motor drive is desired.

The application of a gas engine or gas turbine drive to a centrifugal machine is particularly attractive when the engine or turbine exhaust gases can generate steam in a waste heat boiler to produce additional refrigeration from absorption machine equipment.

Hermetic centrifugal machines are single purpose machines and are generally used for water chilling applications. They are low in first cost because they are a factory package. They can be installed easily and quickly with a minimum of field problems involving motor mounting, coupling and alignment.

STANDARDS AND CODES

Equipment installation should conform to all codes, laws and regulations applying at the site.

The equipment should be manufactured to conform to the ASA B9.1 Safety Code for Mechanical Refrigeration. This safety code requires conformance to the ASME Unfired Pressure Vessel Code.

Specifications should call for conformance to these standards and codes to assure a high quality product. Pressure vessels are ASME stamped when required by the code.

UNIT SELECTION

The factors involved in the selection of a centrifugal machine are load, chilled water or brine quantity, temperature of the chilled water or brine, condensing medium to be used, quantity of the condensing medium and its temperature, type and quantity of power available, fouling factor allowance, amount of usable space available, and the nature of the load, whether variable or constant.

The final selection is usually based on the least expensive combination of machine and heat rejection device as well as a reasonable machine operating cost.

Load, chilled water or brine quantity, and temperature rise are all related to each other such that, when any two are known, the third can be found by the formula:

$$\text{Load (tons)} = \frac{\text{quantity (gpm)} \times \text{temp rise (F)} \times \text{sp ht} \times \text{sp gr}}{24}$$

where:
sp ht = specific heat (1.0 for water)
sp gr = specific gravity (1.0 for water)

Table 7 illustrates typical hermetic centrifugal machine chilled water ratings. Ratings in tons based on various leaving chilled and condenser water temperatures are given for a particular machine size. The ratings in bold face type require rated kilowatt input while those in italics require less input.

Brine cooling normally requires special selection by the manufacturer.

The choice of a chilled water temperature for air conditioning applications should be carefully considered as pointed out in *Part 6*. The selection is an economic one since it involves the analysis of the owning and operating costs of several systems to determine the optimum chilled water temperature

The selection of multiple machines for a common load is normally based on availability, reliability and/or flexibility: availability because of limitations to the physical size it is economical to produce; reliability because of the need to handle a portion of the load when one machine may be down for service; flexibility because of the ability to more efficiently match compressor capacity to partial load requirements. As a general rule, seldom are multiple machine applications made on normal air condition loads less than about 400 tons.

When multiple machines are considered, series water flow thru the coolers may be advantageous (*Fig. 17*). Generally, the longer the piping distribution system, the higher the over-all chilled water rise. For instance, close-coupled chilled water coils and coolers normally have an economic optimum rise of about 8-10 degrees. Conversely, chilled water distribution systems for a campus type operation would normally have an economic optimum rise of about 15-20 degrees. For the higher rises, series water flow thru the chillers may offer an operating cost savings. The first machine operates at a higher suction temperature which requires less power.

TABLE 7—TYPICAL HERMETIC CENTRIFUGAL MACHINE RATINGS

REFRIGERATION CAPACITY (TONS)

Italic Rating Requires Less than 330 kw Input

Leaving Chilled Water Temp (F)	Leaving Condenser Water Temperature (F)			
	85	90	95	100
40	*409*	*405*	*399*	*386*
41	*417*	*414*	406	391
42	*426*	*422*	413	396
43	*434*	429	419	401
44	*442*	435	424	406
45	450	441	430	412
46	*457*	447	435	417
47	*461*	453	442	423
48	*465*	459	449	429
49	*470*	465	456	435
50	*474*	472	462	442

NOTE: Ratings are based on a 2-pass cooler using 380-1260 gpm and on a 2-pass condenser using 430-1430 gpm.

CAPACITY ADJUSTMENT

Water Flow (gpm)		Pass	Nominal Capacity Adjustment to 2-Pass Rating	
Cooler	190 to 630	4	ADD	3%
	250 to 840	3	ADD	1½%
	755 to 2520	1	DEDUCT	4%
Condenser	215 to 715	4	ADD	3%
	285 to 955	3	ADD	1½%
	860 to 2860	1	DEDUCT	2½%

The optimum machine selection involves matching the correct machine and cooling tower as well as the correct entering chilled water temperature and water rise. A selection of several machines and cooling towers often results in finding one combination having a minimum first cost. In many instances it is possible to reduce the condenser water quantity and increase the leaving condenser water temperature, resulting in a smaller tower.

The use of an economizer can effect a power reduction for the compressor of as much as 6% for the same cooler and condenser surface. This same power reduction can be obtained by adding 15% to 30% more surface in the heat exchangers. The method of saving this power is a machine design consideration and becomes a matter for the manufacturer to decide which is the least expensive way to accomplish this reduction. Economizers can only be used with multi-stage compressors.

On low temperature, industrial applications where four or more stages are used, a two-stage economizer is generally justified.

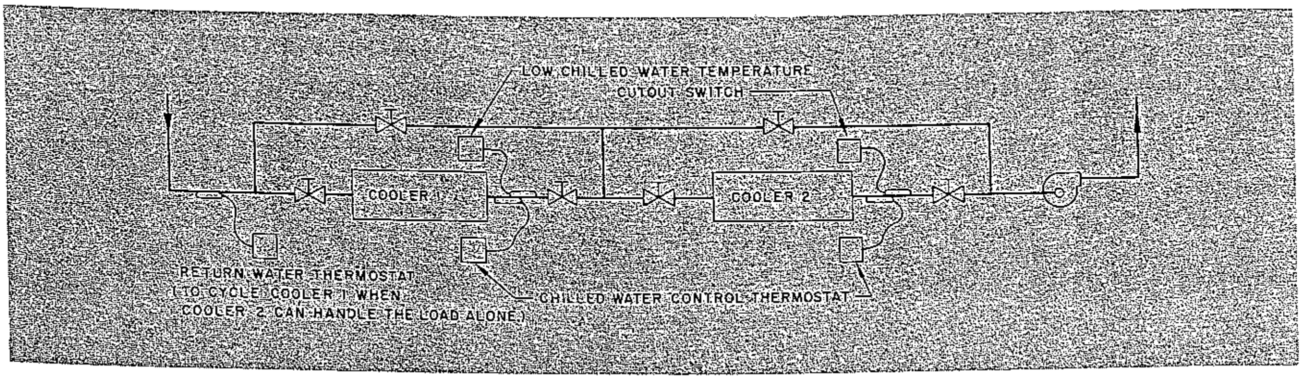

LOW CHILLED WATER TEMPERATURE
CUTOUT SWITCH
COOLER 1
COOLER 2
RETURN WATER THERMOSTAT
(TO CYCLE COOLER 1 WHEN
COOLER 2 CAN HANDLE THE LOAD ALONE.)
CHILLED WATER CONTROL THERMOSTAT

Fig. 17 — Series Arrangement of Two Coolers

Fouling factors used when the cooler and condenser are selected have a direct bearing on the system economics. Too conservative a factor results in a high first cost while too low a factor increases operating costs by increasing the frequency of tube cleaning or increasing the costs of the water conditioning to maintain the low factor. *Part 5* includes detailed effects of fouling on equipment selection, and suggests various fouling factors based on equipment and systems.

Table 8 lists the relative costs and resistance to corrosion of various metals and alloys for special cases where unusual water conditions require other than the standard copper tubing for either the cooler or the condenser.

OPERATING COSTS

The power operating costs of an electrically driven centrifugal machine may be realistically determined by a mechanical integration of power cost increments for the total operating hours of the machine as follows:

$$C = c\,(p_1h_1 + p_2h_2 + p_3h_3 + \qquad + p_nh_n)$$

where:

C = annual power costs

c = cost per kilowatt hour, including demand and energy charges

p = power consumption for incremental percentage of nominal full load, expressed as

(1) motor kilowatt input, OR

(2) $\dfrac{\text{motor output brake horsepower} \times .746}{\text{motor efficiency}}$

(See *Chart 2* for a typical hermetic centrifugal performance.)

h = hours of machine operation during the year at above percentage of nominal full load. (See *Chart 3* for a typical graph showing percentages of full load vs operating hours.)

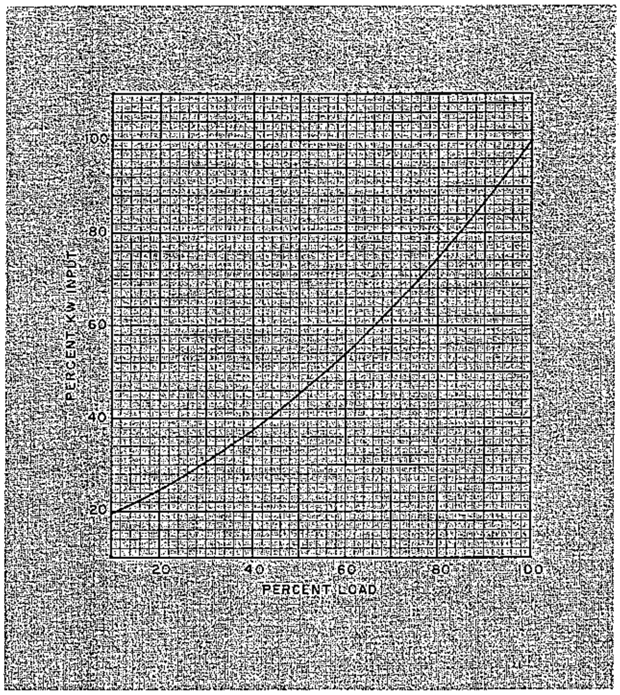
TABLE 8—RELATIVE COSTS AND RESISTANCE TO CORROSION OF METALS AND ALLOYS

		→ CORROSION TENDENCY DECREASES			
Material	Relative Costs per Tube	Sea Water or Brackish Salt Water	Soft Fresh Water high oxygen and carbon dioxide content	Soft Fresh Water low oxygen and carbon dioxide content	Water of High Hardness Content tendency to scale
Steel (SAE 1010)	1.6	N.R.	N.R.	A	A
Copper	1.0	N.R.	B	A	A
Nickel	—	A	A	A	A
Red Brass	1.9	N.R.	B	A	A
Admiralty Brass (inhibited)	2.0	B	A	A	A
Cupro-Nickel 70/30	2.6	A	A	A	A
90/10	1.9	A	A	A	A
Aluminum Brass	2.5	A	A	A	A
Stainless Steel (304L)	6.0	N.R.	A	A	A
Nickel Steel (3½%)	2.6	N.R.	N.R.	A	A
Aluminum	—	N.R.	B	B	B

NOTE: The relative resistance to corrosion from fresh water increases as the tendency for scaling increases. Water that causes scaling usually does not cause corrosion.

Symbols in Table: A — Generally acceptable for use.
B — Used under certain conditions, when experience shows metal is acceptable.
N.R — Not recommended.

CHART 2—TYPICAL HERMETIC CENTRIFUGAL PERFORMANCE



The annual power costs of auxiliary equipment may be calculated as follows:

$$\text{Power costs} = \frac{.746 \times \text{bhp} \times \text{hr} \times \text{cost/kw-hr}}{\text{motor efficiency}}$$

DRIVE SELECTION

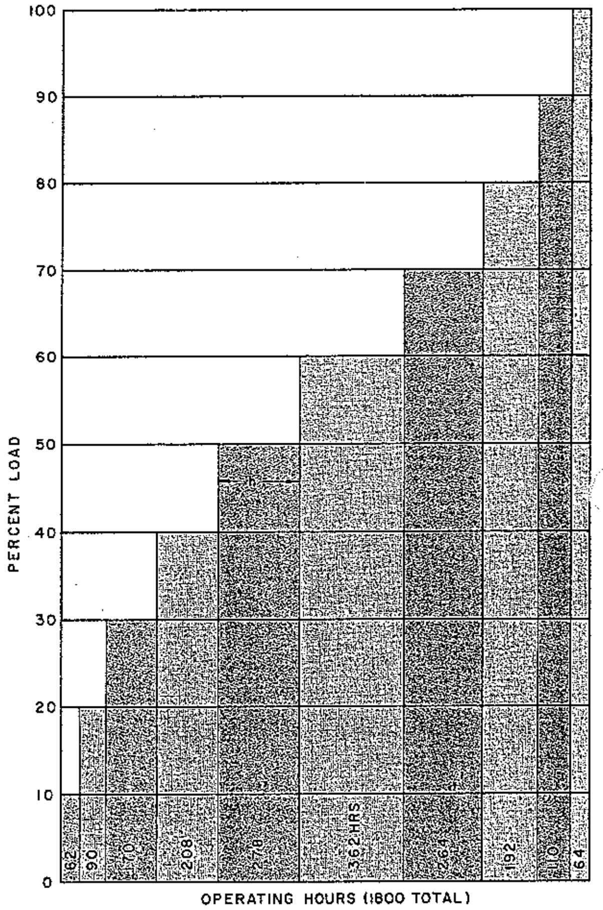
There are four types of drives in general use for centrifugal compressors.

1. Steam turbine
2. Variable speed motor
3. Constant speed motor
4. Constant speed engine

Steam turbines are ideally suited for centrifugal compressors. They afford variable rpm, permitting the compressor to operate at a minimum speed and brake horsepower. They usually have a good efficiency characteristic over the required speed range with economy of operation. Refer to *Part 8* for additional information on drives

Variable speed motors of the wound rotor type are used for open centrifugal machine applications because of the favorable starting inrush characteristics and the range of speed regulation. Capacity can be controlled by varying the speed manually or automatically. *Figure 18* indicates that a rapid decrease in power input results when the speed is reduced

CHART 3—TYPICAL GRAPH, PERCENT FULL LOAD VS OPERATING HOURS



Motors normally used for a constant speed drive are the squirrel cage induction or synchronous type. It is sometimes possible to obtain a motor that has low starting current features and uses an across-the-line starter, thus saving on starting equipment co when power company limitations on inrush current are satisfied.

Hermetic compressors use only induction type motors since they usually operate in a refrigerant atmosphere and do not require brushes or commutators which may cause a breakdown of the refrigerant due to arcing.

Synchronous motors may be applied to advantage if a power factor correction is desired. Another method is to use a standard induction motor plus the necessary capacitor.

Natural gas engines may also be applied as drives for centrifugal machines. Engine speeds normally range from 900-1200 rpm, the lower speeds being used on applications having longer annual operating hours. Speed increasing gears are used between the engine and the centrifugal machine.

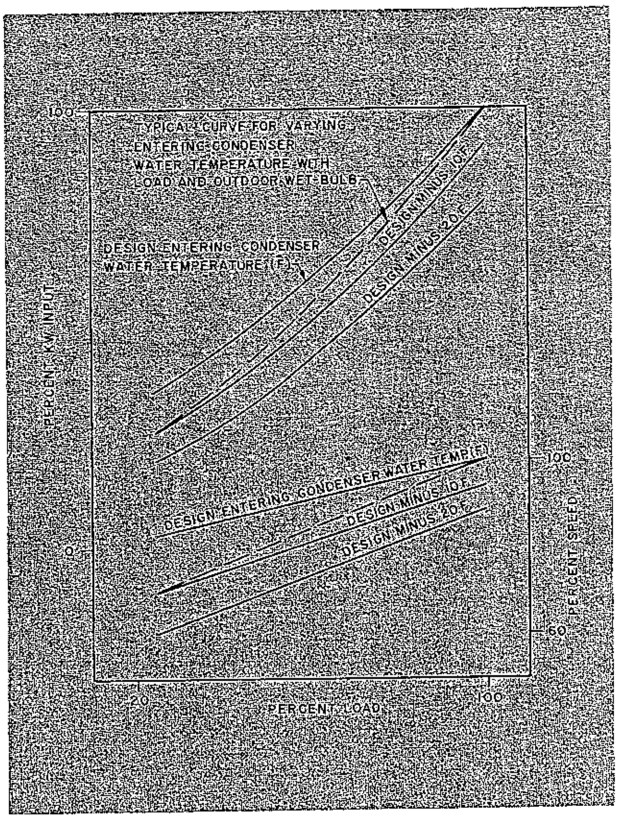


Fig. 18 — Typical Machine Performance, Wound Rotor Motor

Centrifugal compressors have low starting torques; therefore, most drives can easily be matched with these machines.

However, not only must starting torque be checked, but also acceleration time required to bring the centrifugal up to speed. Too fast an acceleration time is not desirable because design stresses for keyways may be exceeded and lubrication problems may be created. Minimum recommended acceleration times for open machines are available from the manufacturer.

GEARS

Speed increasing gears used for an open centrifugal compressor drive are usually the double helical (herring-bone) type The horsepower loss in the gear must be included with the actual compressor brake horsepower to determine the motor horsepower.

The selection of the proper gear for a particular centrifugal application depends on motor horsepower, motor rpm and compressor rpm. Water-cooled oil coolers are normally included with the gear.

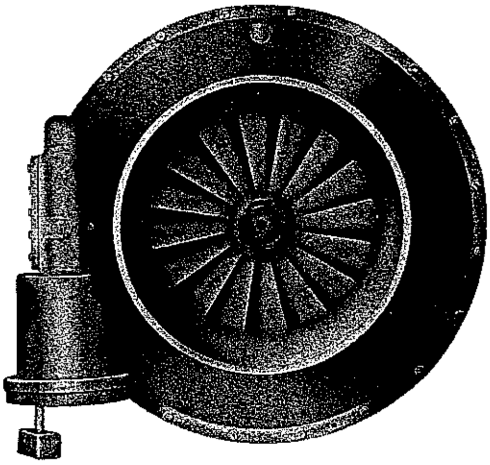


Fig. 19 — Typical Variable Inlet Guide Vanes

MOTOR STARTING EQUIPMENT

Both hermetic and open centrifugals may require the more commonly applied starters. These are discussed in *Part 8* and include across-the-line, star-delta, primary resistance, auto transformer, and primary reactor starters

CONTROLS

CAPACITY CONTROL

If a centrifugal machine is to perform satisfactorily under a partial load, a means of effecting a capacity reduction in proportion to the reduction of the instantaneous load is required.

Hermetic Centrifugal

Water temperature control is obtained by means of variable inlet guide vanes (*Fig. 19*) at the suction inlet to the compressor.

This control reduces capacity by varying the angle at which the suction gas is directed into the eye of the impeller. It also conserves power because it promotes aerodynamic gas flow thru the compressor. At low flow the change of inlet gas direction has little effect on capacity and the control operates primarily as a suction damper. The minimum partial load capacity of the machine is based upon the amount of gas leakage thru the fully closed capacity regulating vanes. *Chart 2* shows a typical power input curve for a hermetic centrifugal machine operating with condenser water supplied from a cooling tower when the refrigeration load closely follows the outdoor wet-bulb temperature. The curve is based on the design water flow being maintained at a constant rate for both the cooler and condenser.

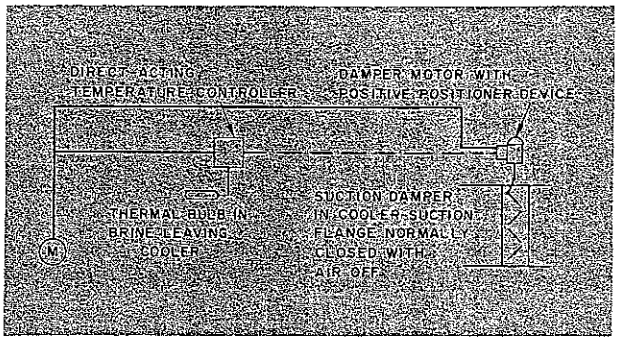

DIRECT ACTING TEMPERATURE CONTROLLER
DAMPER MOTOR WITH POSITIVE POSITIONER DEVICE
THERMAL BULB IN BRINE LEAVING COOLER
SUCTION DAMPER IN COOLER SUCTION FLANGE NORMALLY CLOSED WITH AIR OFF
M

FIG. 20 — SUCTION DAMPER CONTROL SYSTEM

A chilled water control thermostat automatically controls the leaving chilled water temperature. When the temperature changes, the thermostat signals the chilled water control to reposition the capacity regulating vanes which change the capacity of the machine to maintain the desired temperature. When the vanes reach the closed position and the leaving temperature continues to decrease to a predetermined minimum, the low chilled water temperature cutout switch stops the machine.

Open Centrifugal

Capacity control on an open centrifugal machine may be obtained with a suction damper (*Fig. 20*), variable inlet guide vanes, or variable speed drive (steam turbine, gas turbine, gas engine or wound rotor motor).

A suction damper is controlled by a thermostat in the leaving chilled water to reduce the capacity of the compressor by throttling the suction gas. The variable inlet guide vane control is identical to that discussed under *Hermetic Centrifugal*.

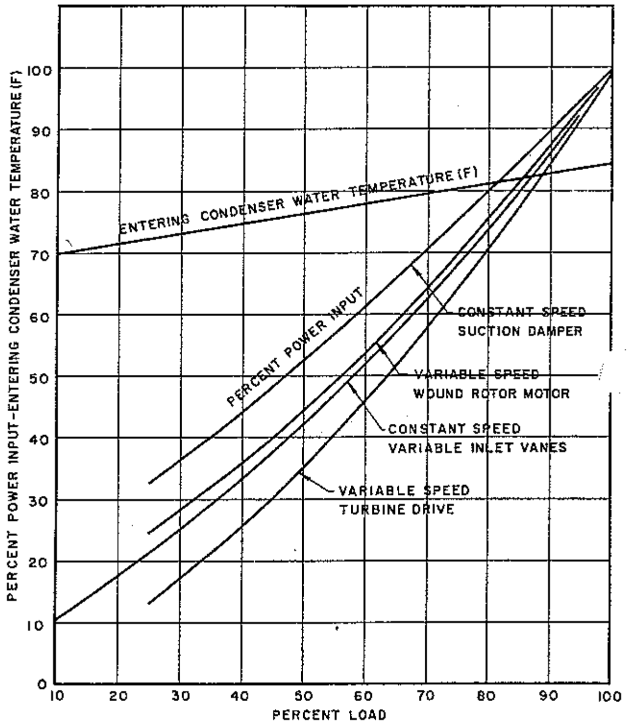
Variable speed drives may be controlled manually when the change in loading is gradual or when a suction damper is used for automatic control. Automatic speed control is used with steam turbine, gas turbine or gas engine drives. Automatic speed control provides very economical operation and requires less input than other methods of control. Automatic speed control of wound rotor motors is expensive and is seldom used.

Chart 4 shows the comparative performances of different methods of centrifugal compressor capacity control.

CONTROL OF SURGE

Surge is a characteristic of centrifugal compressors which occurs at reduced capacities. This condition is a result of the breakdown in flow which occurs in the impeller. When this happens, the impeller can no longer maintain the condenser pressure, and there occurs a momentary reversal of flow which is accompanied by a lowering of condenser pressure. This allows the impeller to function normally again, and the gas flow returns to its normal direction. The operation is stable until the condenser pressure builds up and surge occurs again. Surging can be detected primarily by the change in soun level of the machine.

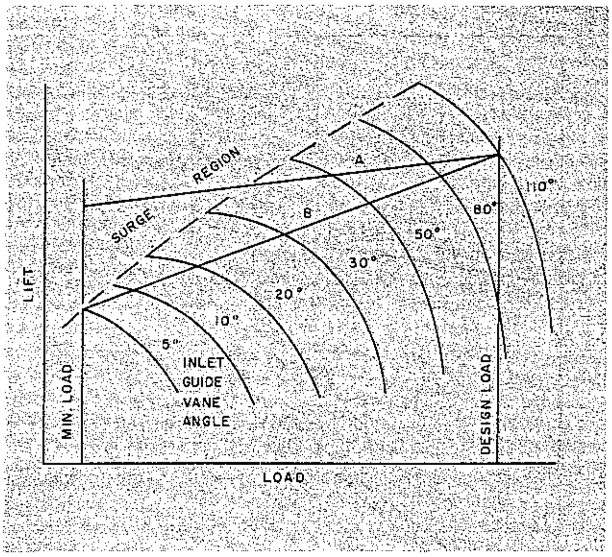
CHART 4—COMPARATIVE PERFORMANCES, CENTRIFUGAL COMPRESSOR CAPACITY CONTROL



Surge in a centrifugal machine does not occur at partial loads if the head or lift decreases sufficiently with the load.

Chart 5 shows a typical lift versus load diagram for a hermetic centrifugal. A series of curves is plotted indicating the compressor operating curves at different positions of the inlet guide vanes. Line *B* represents a series of operating points of a machine when the characteristic of the loading is such that the condensing temperature or the total lift of the compressor reduces as the loading also reduces. An example is a comfort air conditioning job using a cooling tower to provide the condenser water. As the outdoor wet-bulb decreases, the refrigeration load decreases and the condenser water temperature is reduced, allowing the condensing temperature to drop. Line *A* represents a series of operating points of a machine when the condensing temperature or lift remains almost constant or decreases only

CHART 5—TYPICAL LIFT-LOAD DIAGRAM, HERMETIC CENTRIFUGAL



slightly. An example is the condition in which a fixed temperature of condenser water is available all year, or the condenser water temperature is still at the design temperature and a partial load condition exists as in a process application.

It can be seen from the chart that the line *B* does not enter the surge region until the loading is under the minimum load (approximately 10%). The line *A* enters the surge region above the minimum loading and, therefore, the machine needs some method of maintaining the loading above this point, such as a hot gas bypass.

To control surge occurring at partial load for either an open or hermetic centrifugal machine, a valved gas connection between the condenser and cooler is normally used to load the compressor artificially. The valve may be either manual or automatic (*Fig. 21*). As applied to open centrifugal machines, the automatic hot gas bypass valve is usually controlled in sequence with the automatic suction damper or with the speed of the compressor so that the valve starts to open just before the suction damper position or speed of the compressor indicates surge.

SAFETY CONTROL

Hermetic Centrifugal

The variable inlet guide vanes control capacity and are used to prevent motor overload in two ways:

1. When starting, the capacity regulating vanes remain closed until the motor is connected across the line at full voltage and the current drawn is below full load current

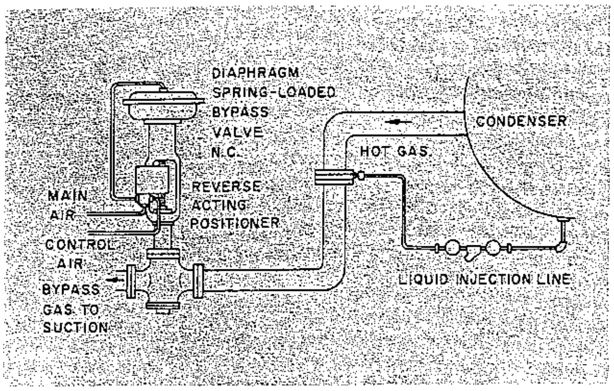


FIG. 21 — AUTOMATIC HOT GAS BYPASS VALVE

2. Motor overload control overrides the chilled water temperature control to prevent further opening of the vanes at 100% motor load. If the current drawn continues to rise above 100%, the vanes begin to close, reducing the motor load.

Similar controls may be obtained for an open centrifugal machine driven by a constant speed motor.

A typical ladder diagram of the safety controls for a hermetic centrifugal is shown in *Fig. 22*. Safety controls common to both the hermetic and open type of centrifugal machine are described as follows:

1. Condenser high pressure cutout switch stops the compressor when the condenser pressure becomes too high due to a condenser water stoppage, excessive condenser scaling or air in the system.
2. Low refrigerant temperature cutout switch stops the compressor when the evaporator pressure becomes too low due to a chilled water stoppage, excessive cooler scaling, or insufficient refrigerant charge.
3. Low oil pressure cutout switch stops the compressor when the oil pressure drops below the required minimum and prevents either starting (on compressors with external oil pumps) or operating (on compressors with shaft-driven oil pumps) the compressor motor before the oil pressure is up to the minimum.
4. Low brine or chilled water temperature cutout switch stops the compressor when the leaving brine or chilled water temperature drops below the minimum allowable temperature.
5. Chilled water flow switch stops the compressor when chilled water ceases to flow, and prevents a start-up of the compressor motor until chilled water flow is established (optional).

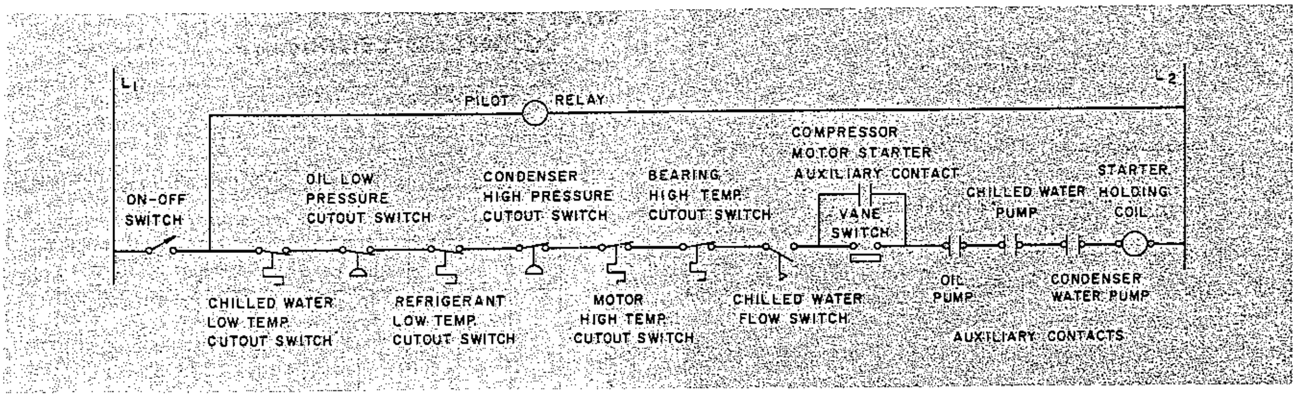

L1
L2
PILOT RELAY
ON-OFF SWITCH
OIL LOW PRESSURE CUTOUT SWITCH
CONDENSER HIGH PRESSURE CUTOUT SWITCH
BEARING HIGH TEMP CUTOUT SWITCH
COMPRESSOR MOTOR STARTER AUXILIARY CONTACT
VANE SWITCH
CHILLED WATER PUMP
STARTER HOLDING COIL
CHILLED WATER LOW TEMP CUTOUT SWITCH
REFRIGERANT LOW TEMP CUTOUT SWITCH
MOTOR HIGH TEMP CUTOUT SWITCH
CHILLED WATER FLOW SWITCH
OIL PUMP
CONDENSER WATER PUMP
AUXILIARY CONTACTS

Fig. 22 — Typical Safety Control System, Hermetic Machine

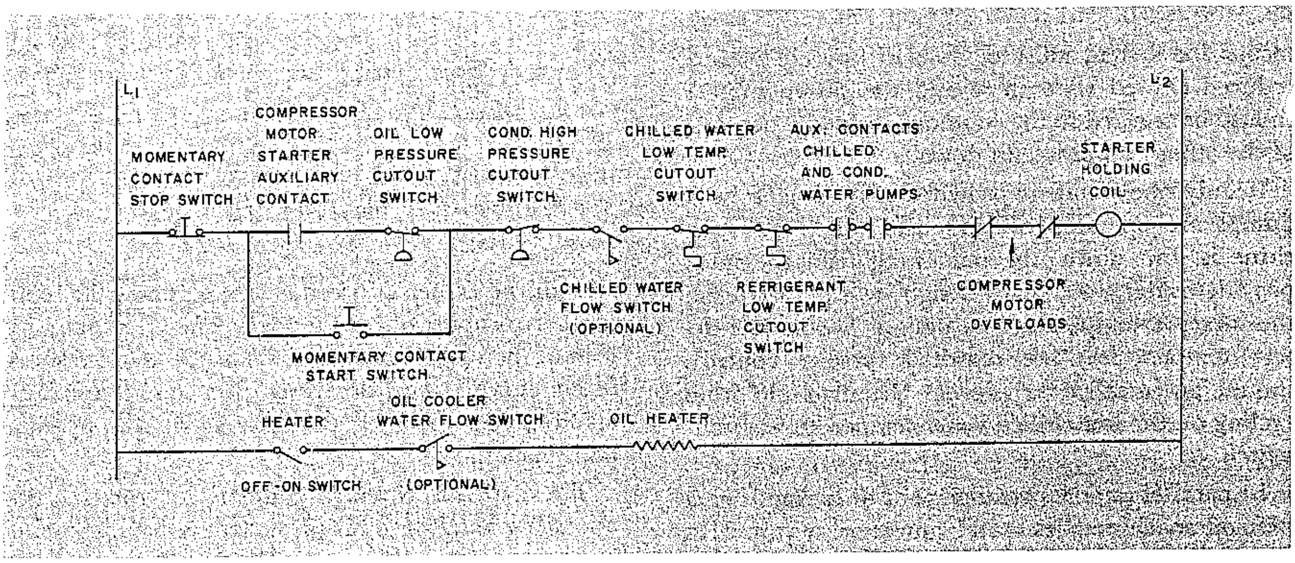

L1
L2
MOMENTARY CONTACT STOP SWITCH
COMPRESSOR MOTOR STARTER AUXILIARY CONTACT
OIL LOW PRESSURE CUTOUT SWITCH
COND. HIGH PRESSURE CUTOUT SWITCH
CHILLED WATER LOW TEMP CUTOUT SWITCH
AUX. CONTACTS CHILLED AND COND. WATER PUMPS
STARTER HOLDING COIL
CHILLED WATER FLOW SWITCH (OPTIONAL)
REFRIGERANT LOW TEMP CUTOUT SWITCH
COMPRESSOR MOTOR OVERLOADS
MOMENTARY CONTACT START SWITCH
HEATER
OIL COOLER WATER FLOW SWITCH
OIL HEATER
OFF-ON SWITCH
(OPTIONAL)

Fig. 23 — Typical Safety Control System, Open Motor-Driven Machine

Open Centrifugal

A ladder diagram of the safety controls for a motor-driven centrifugal is shown in *Fig. 23*.

ELECTRICAL DEMAND CONTROL

This control can override the capacity control to limit the current drawn during off-season operation. This allows operation of the machine without creating high electrical demand charges during months when full load capacity is not required. The control can be set to reduce the amount of current which can be drawn by the motor down to as low as 40% of the full load amperage.

MULTIPLE MACHINE CONTROL

When two or more centrifugal machines are required to handle a load, they may be applied in a parallel or series arrangement of coolers. The arrangements are controlled in a manner similar to single machines.

Installations with machine coolers in parallel may utilize two or more machines. With series chilled water flow, the cooler pressure loss is cumulative and may become excessive if more than two machines are installed in series.

Parallel Arrangement

When two or more machines are installed with the coolers connected in parallel in the chilled water circuit (*Fig. 24*), each machine may control its own leaving chilled water at design temperature as in a single machine installation. The same throttling range should be used for each machine. As the system load is reduced, each machine reduces capacity simultaneously, thus individually producing the same leaving chilled water temperature.

When each cooler is provided with a separate chilled water pump, the pump and cooler may be shut down during partial load operation. This

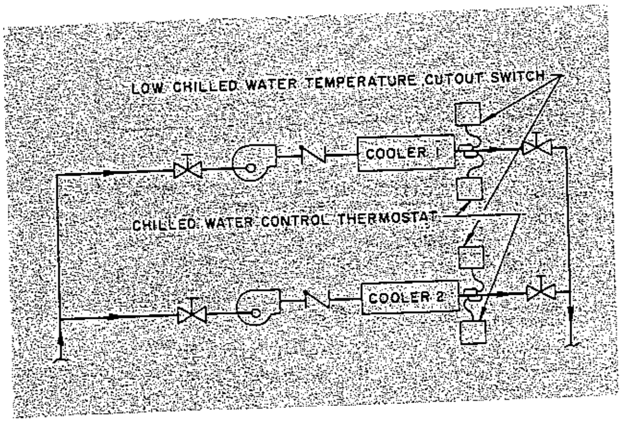

LOW CHILLED WATER TEMPERATURE CUTOUT SWITCH
COOLER 1
CHILLED WATER CONTROL THERMOSTAT
COOLER 2

FIG. 24 – PARALLEL ARRANGEMENT OF TWO MACHINES (TWO PUMPS)

means the system must be able to operate with a reduced chilled water flow and the pump motors should be selected so they do not overload when one of the other pumps is shut down.

If only one pump is provided (*Fig 25*) or both pumps are operated continuously, when one machine is shut down, the remaining machine must provide colder water than design in order to bring the mixture temperature to design. When low design temperatures are required, proper controls should be installed to prevent the machine from cycling on the low chilled water temperature cutout switch

In parallel or series arrangement of hermetic centrifugal machines, less total power is required to operate both machines simultaneously down to approximately 35% load than to run only one, throttling it to the load. This occurs because the surface area in the cooler and condenser is greater at light loads in proportion to the load. The effect may be seen in the shape of the load versus percent kilowatt input curve (*Chart 6*) Note that above approximately a 35% load, less power is required to operate both machines.

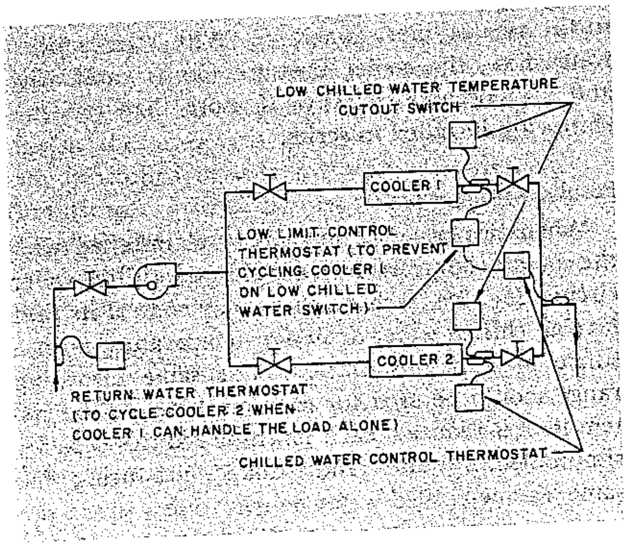
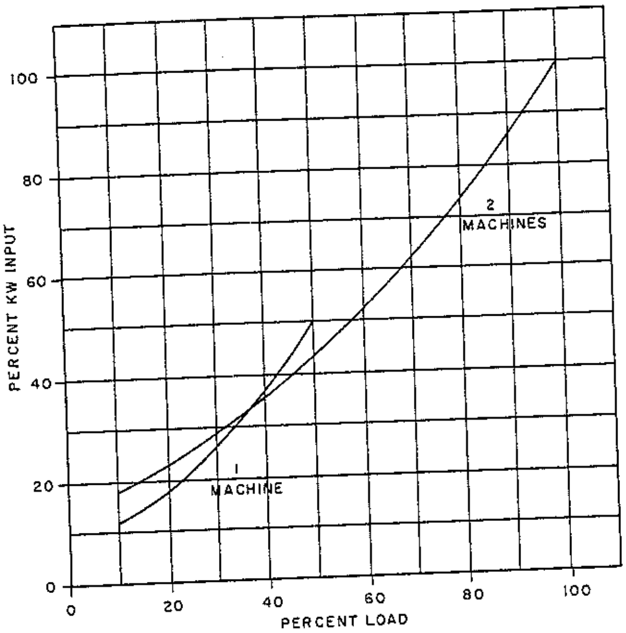

LOW CHILLED WATER TEMPERATURE CUTOUT SWITCH
COOLER 1
LOW LIMIT CONTROL THERMOSTAT (TO PREVENT CYCLING COOLER 1 ON LOW CHILLED WATER SWITCH)
COOLER 2
RETURN WATER THERMOSTAT (TO CYCLE COOLER 2 WHEN COOLER 1 CAN HANDLE THE LOAD ALONE)
CHILLED WATER CONTROL THERMOSTAT

FIG. 25 – PARALLEL ARRANGEMENT OF TWO MACHINES (ONE PUMP)

CHART 6—TYPICAL PERFORMANCE, PARALLEL CONNECTED CENTRIFUGAL MACHINES



The expense of extra controls to equalize the operating time of multiple machine arrangements is not required. When using reciprocating equipment, changing the order of starting and stopping multiple compressor arrangements is sometimes justified. Due to the absence of wearing parts in a centrifugal machine, this changing is seldom used.

Series Arrangement

When coolers are connected in series, equal reduction of loading of each machine produces the best power consumption. The throttling range of the high stage machine must be adjusted to insure that each machine handles the same percentage of the system load, both at design and at partial load conditions.

In any series selection the throttling range required on the high stage machine equals the chilled water temperature drop thru the low stage machine plus the throttling range of the low stage machine.

Figure 17 shows a series cooler arrangement of two hermetic centrifugal machines and controls The extra thermostat (return water thermostat) is used to cut in and cut out the first machine at light loads.

TABLE 9—NORMAL REFRIGERANT LOSS EXPECTED WITH CENTRIFUGAL MACHINES

Installation Size (tons)	125-175	175-250	250-350	350 or more
Refrigerant Loss (lb/yr)	75-125	100-150	100-150	125-175

*Based on comfort conditioning 120 days per year, 10 hours per day (1200 hour season).

NOTE: Factors for conditions other than above:
Time factors — 2500 hr/yr 1 25
— continuous for year 1.50
Application factor, low temp 1.20

PURGE UNIT

A purge unit for a centrifugal machine may be either a thermal or a compressor type.

The purpose of the unit is to evacuate air and water from the centrifugal machine and to recover and return refrigerant which is mixed with the air. Even though a machine may be perfectly airtight, it may develop a water leak which is detected only by operation of the purge system. If water is allowed to remain in the machine, serious damage to tubes and other internal parts can result.

The compressor type purge unit operates independently by means of a small reciprocating compressor to remove the air, moisture and a small quantity of refrigerant from the condenser.

The thermal type purge unit operates on a pressure differential principle, and a reciprocating compressor is not required.

Although the purge unit performs a highly efficient job of removing refrigerant from the air being purged, it is physically impossible to recover all the entrained refrigerant; some is always lost.

Table 9 shows an approximation of the normal refrigerant loss to be expected with a centrifugal machine. It must be realized that the actual loss varies widely from one installation to the next; this variation is based on machine tightness, frequency of purging and other factors.

INSULATION

The cooler, suction piping and other cold surfaces should be insulated to prevent sweating. Float valve chambers, water boxes and other parts of the machine which may require servicing should be provided with a removable type of insulation, such as sheet metal covers filled with granulated cork.

Various types of insulation can be used, such as vegetable cork, closed cell foamed plastic and expanded polystyrenes.

LOCATION

Machine location and layout should be carefully studied when applying a centrifugal machine. The location of the machine directly influences the economic and possibly the sound level aspects of any system.

The construction of the room where the machine is located should contain mass to reduce transmissibility of noise to surrounding spaces and should also provide acoustical treatment to maintain reasonable sound levels in the room.

A floor adequately strong and reasonably level is all that is required for the location of a hermetic centrifugal machine. However, though the foregoing statement may be taken literally, it is to the engineer's advantage to consider other aspects relative to the location of the machine.

1. It should be located so that the installation costs of the piping between the unit and the equipment it supplies and the costs of the wiring or piping of the services to the unit are at a minimum.
2. There should be sufficient space near the machine for auxiliary equipment such as chilled and condenser water pumps and piping.
3. There should be adequate clearances around the machine for access and servicing.

In new construction on upper floors, steel floor framing should be laid out by the architect or consulting engineer to match machine supports in order to transfer loads to the building columns. On upper floors in existing buildings, the use of existing floor slabs should be avoided. Supplementary steel framing for transferring all machine loads to building columns should be designed by the structural engineer.

LAYOUT

In the layout of centrifugal refrigeration machines, consideration should be given to nozzle arrangements.

NOZZLE ARRANGEMENTS

Cooler — When arranged for multi-pass, water should enter the bottom tubes and leave thru the top tubes. This method gives the best efficiency and promotes venting of any air trapped inside the tubes.

Condenser — When arranged for multi-pass, water should enter the top tubes first. This provides the coldest surface at the top of the condenser shell to stratify noncondensables for proper purging.

The nozzle arrangement chosen for both cooler and condenser should result in a minimum number of chilled water and condenser water pipe fittings, the optimum access to the centrifugal and auxiliary equipment, and a neat appearance.

OUTDOOR INSTALLATION

A hermetic centrifugal machine is basically designed for indoor operation. Outdoor installation is usually not encouraged. The machines should not be located outdoors when they may be subjected to freezing temperatures.

A simple heated structure enclosing the machine is preferred since complete protection for the machine, instruments, starter and auxiliary equipment is provided. Erection of such an enclosure may be less costly than the outdoor installation precautions which may be required. If it is necessary to install the machine outdoors, refer to the manufacturer for recommendations and precautions

UNIT ISOLATION

Normally, only the hermetic compressor assembly is isolated from the floor with moulded grooved neoprene isolation pads. For upper story installations, isolation pads may also be required under the feet of the cooler. In the case of highly critical installations, spring isolators may be required under the compressor assembly and cooler, in which case auxiliary pumps and piping should be isolated

The base which holds the open compressor and its driver is designed for each individual job. With a steel base design, the complete unit is mounted on an independent fabricated steel foundation. With a concrete type of base, the various components are mounted on individual steel plates which are anchored to the concrete.

Figure 26 shows a depressed concrete base isolated from the adjoining floor with mastic.

Cork is not a satisfactory isolation material for most applications. Under no conditions should it be considered for isolation on an upper floor of a building where the least amount of vibration would be objectionable. However, four-inch thick cork pads may be used in noncritical locations.

The machine foundation should be located away from building column footings. In order to do a first class job such as may be required on an upper floor, spring mounting should be used. Sandwich type spring mountings, although nonadjustable, may also be used under a metal pan into which the concrete is poured.

When spring isolation is used, flexible rubber connections are recommended at the points where the chilled and condenser water piping is connected to the cooler and condenser to take up the movement of machine and base when starting and stopping.

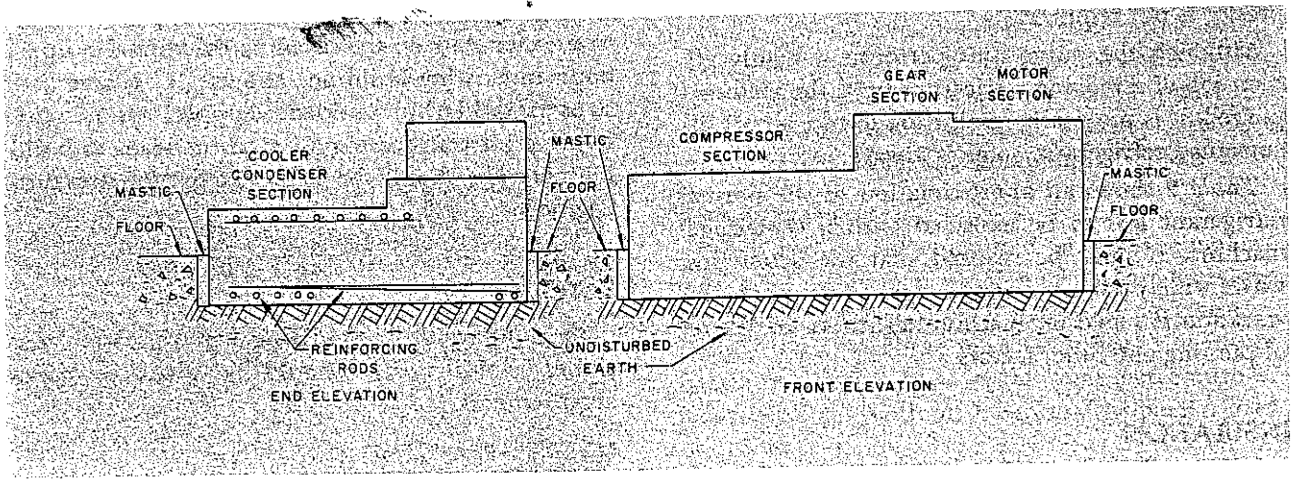


FIG. 26 — DEPRESSED CONCRETE BASE (ISOLATED FROM FLOOR)

CHAPTER 3. ABSORPTION REFRIGERATION MACHINE

The absorption refrigeration machine is a water chilling package which uses heat directly without the use of a prime mover, thus utilizing the heating facilities on a full time, year-round basis. Because of its compactness and vibrationless operation, it can be installed anywhere space and a heat source is available, from basement to roof. It uses the cheapest, safest and most available of all refrigerants, ordinary tap water Its absorbent is a simple salt.

This chapter presents data to guide the engineer in the practical application and layout of absorption refrigeration equipment when used for comfort air conditioning systems.

APPLICATION

Since heat in the form of steam or hot water is generally the operating force of an absorption machine, the following situations are favorable to the application of absorption refrigeration machines:

1. Where low cost fuel is available, as in natural gas regions.
2. Where electric rates are high. Whenever the cost of steam in dollars per thousand pounds is less than fifty times the cost of electricity in dollars per kilowatt, a lower operating cost can be expected for the absorption machine. This is approximately the break-even point in operating cost (at design) between this machine and the electrically-driven compressor The cost of steam is shown graphically in *Chart 7* for different fuels. Comparison curves in *Chart 8* indicate the operating cost of refrigeration for various costs of steam and electricity when applied to an absorption and centrifugal machine respectively. Demand charges should be included in the average electric cost. When comparing the operating cost of a steam turbine-driven centrifugal and an absorption

CHART 7—COMPARATIVE COSTS OF STEAM

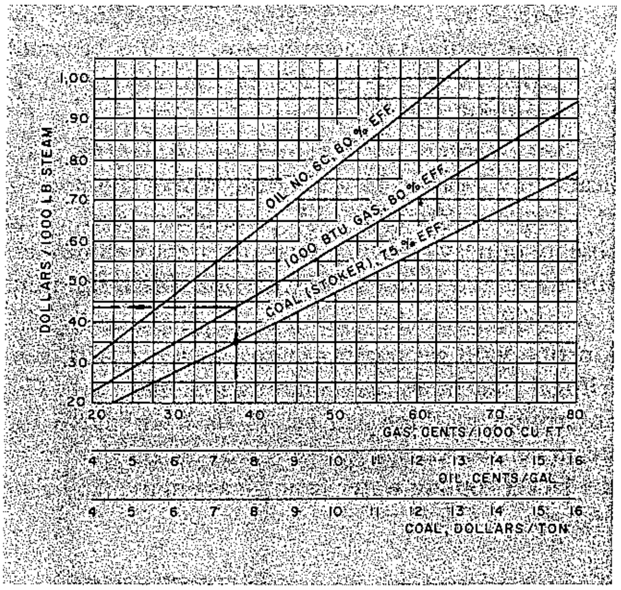
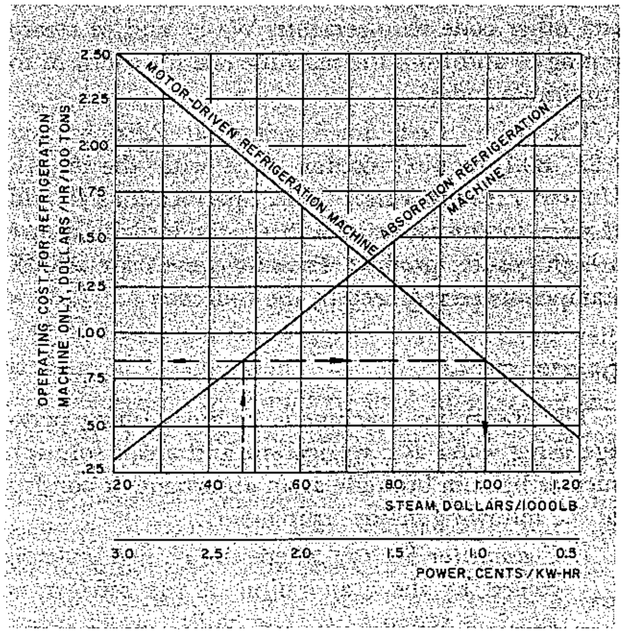


CHART 8—STEAM COSTS VS POWER COSTS



machine, a straight steam rate per ton of refrigeration is not a proper criterion. To obtain a correct analysis, the total system heat input should be used

3. Where steam or gas utilities are desirous of promoting summer loads.
4. Where low pressure heating boiler capacity is largely or wholly unused during the cooling season.
5. When waste steam is available.
6. Where there is a lack of adequate electric facilities for installing a conventional compression machine. Since the absorption machine uses only 2-9% of the electric power required by compression type equipment, its use becomes attractive where emergency stand-by power is required, as in hospitals.

The absorption machine can be installed in practically any location in a building where the floor is of adequate strength and reasonably level. The absence of heavy moving parts practically eliminates vibrations and reduces the noise level to a minimum.

Absorption machines may be applied also in conjunction with gas engines or turbines and with centrifugal machines as combination systems. The absorption machine can use as its heat source the steam or hot water made in a waste heat boiler or the jacket cooling water from a gas engine (250 F or higher).

STANDARDS AND CODES

The location and installation of absorption machines should be made in accordance with local and other code requirements. Water and/or steam piping to and from the machine should conform to applicable codes.

The Safety Code for Mechanical Refrigeration ASA B9.1 requires conformance with the ASME Unfired Pressure Vessel Code. Specifications should require conformance with these standards to assure a high quality product. Pressure vessels are ASME stamped when required by the code.

DESCRIPTION

The absorption machine is a water-chilling package using water as a refrigerant and a salt solution such as lithium bromide as an absorbent. It consists of the following major components:

1. *Evaporator Section* where the chilled water is cooled by the evaporation of the refrigerant which is sprayed over the chilled water tubes.
2. *Absorber Section* where the evaporated water vapor is absorbed by the absorbent. The heat of absorption is removed by condenser water circulated thru this section.
3. *Generator Section* where heat is added in the form of steam or hot water to boil off the refrigerant from the absorbent to reconcentrate the solution.
4. *Condenser Section* where the water vapor produced in the generator is condensed by condenser water circulated thru this section.
5. *Evaporator Pump* which pumps the refrigerant over the tube bundle in the evaporator section.
6. *Solution Pumps* which pump the salt solution to the generator and also to the spray header in the absorber.
7. *Heat Exchanger* where the dilute solution being pumped to the generator from the absorber is heated by the hot concentrated solution which is returned to the absorber.
8. *Purge Unit* which is used to remove noncondensables from the machine and to maintain a low pressure in the machine.

Figure 27 shows a schematic of an absorption cycle. The machine may be constructed in one, two or more shells or sections depending on the manufacturer or the application.

UNIT SELECTION

The factors influencing the selection of an absorption machine are load, chilled water quantity, temperature of the chilled water, condenser water source, condenser water temperature, condenser water quantity, fouling factor allowance, and heat source. The final selection is usually based on the least expensive combination of machine and cooling tower as well as a reasonable machine operating cost. The absorption machine can be utilized with any conventional open or closed circuit chilled water system.

Load, chilled water quantity, and temperature rise are all related to each other so that, when any two are known, the third can be found by the formula:

$$\text{Load (tons)} = \frac{\text{water quantity (gpm)} \times \text{temp rise}}{24}$$

Table 10 illustrates typical absorption machine chilled water ratings using steam as the energy source. Ratings in tons based on various leaving chilled water temperatures, entering condenser

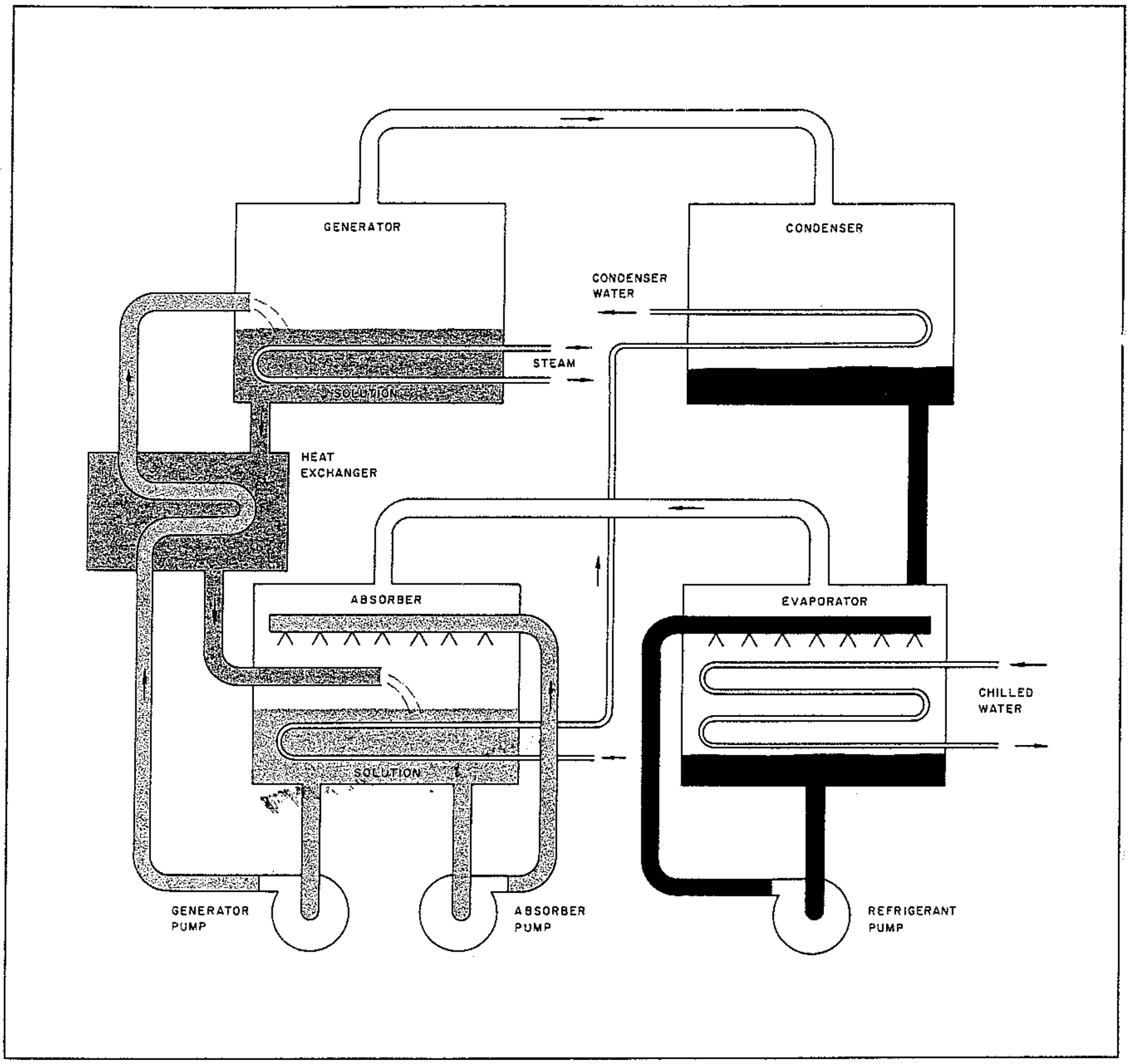


Fig. 27 — Schematic of Basic Absorption Cycle

water temperatures and steam pressures are given for a particular machine size.

The chilled water temperature should be carefully chosen rather than casually selected or assumed. The proper determination of water quantity and temperature for chilled water coils is discussed in *Part 6*. When low water quantities and a high rise (15 to 20 degrees) are required for the chilled water system, the use of two machines piped in series may be an economic advantage since one machine operates at a higher level and requires less heat input.

Almost any source of condenser water is suitable for use in an absorption machine, provided it is of a good quality. Cooling towers are generally used, but river, lake or well water can also be used when available in sufficient quantity and temperature.

If the condenser water source is a lake, river, well or existing process water, the maximum expected water temperature should be used in selecting the machine. The water quantity required depends on the temperature and load. When a cooling tower is to be used in conjunction with the absorption machine, the tower selection should be matched to the machine selection to provide the most economical combination. In many cases the optimum tower selection will indicate a condenser water tempera-

TABLE 10—TYPICAL ABSORPTION MACHINE RATINGS

REFRIGERATION CAPACITY (TONS)

(Italic Ratings Require Less Than Nominal Condenser Water Flow)

LEAVING CHILLED WATER TEMP. (F)	Entering Condenser Water Temperature (F)											
	75			80			85			90		
	STEAM PRESSURE (PSIG)											
	12	10	8	12	10	8	12	10	8	12	10	8
40	766	759	723	707	695	664	639	628	598	545	529	482
42	824	805	773	761	749	714	702	682	648	605	592	551
44	870	856	818	817	797	760	750	737	695	663	643	603
45	889	878	840	838	823	782	773	760	717	688	673	630
46	*889*	*889*	858	858	843	802	800	784	737	710	700	654
48	*889*	*889*	*889*	*889*	883	840	840	825	778	758	745	696
50	*889*	*889*	*889*	*889*	*889*	877	872	857	814	800	782	728
	STEAM PRESSURE (PSIG)											
	6	4	2	6	4	2	6	4	2	6	4	2
40	700	659	631	635	593	556	562	505	454	426	—	—
42	748	711	676	685	637	595	608	557	500	495	411	—
44	791	756	718	730	686	637	652	602	539	553	480	—
45	814	776	739	748	709	662	674	624	559	576	519	385
46	832	797	761	770	731	682	697	647	580	597	539	406
48	871	838	803	808	771	723	736	686	622	636	578	459
50	*889*	876	840	843	811	765	771	725	665	674	610	496

ture higher than the temperature normally estimated, which is usually 7 to 10 degrees above the design wet-bulb temperature. This may mean a considerable saving in cooling tower cost by reducing the size of the tower. Since this is a heat-operated machine, the heat rejection to the cooling tower is approximately two times that of a motor-driven refrigeration machine. The cooling tower used with the absorption machine is usually about 75% larger than that used with the motor-driven machine. The condenser water temperature drop thru the tower is usually about 17 to 20 degrees. Refer to *Chapter 5* for details on the economics of the cooling tower selection.

Typical fouling factor allowances which should be used for the chilled water and condenser water systems in the selection of the machine are given in *Part 5, Water Conditioning*. Generally a *minimum* factor of .0005 is used for both a closed recirculating chilled water system and an open recirculating condenser water system with conditioned water.

Absorption machines normally use either low pressure steam or high temperature water as an energy source. The pressure or temperature limits are usually defined by the manufacturer, although 12 psig steam pressure or a leaving hot water temperature of 240 F and a temperature drop of 160 degrees are usually considered as maximum values. When the temperature or pressure (energy source) exceeds the machine design limits, methods such as a steam pressure reducing valve, a steam-to-hot water converter, a hot water-to-steam converter, a water-to-water heat exchanger, or a run-around system, blending return water with supply water can be used to reduce the energy source to the acceptable limits. Other energy sources that may be adapted for use in the absorption machine are hot chemical solutions or petroleum.

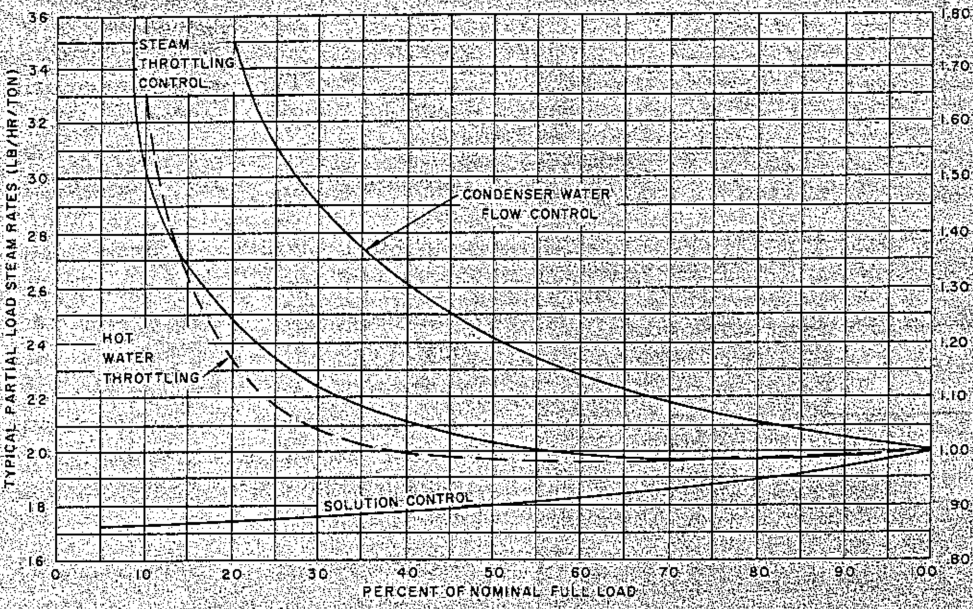
Whenever the capacity requirements are less than the capacity of the machine, a lower operating steam pressure should be considered. This usually permits a lower steam rate and a lower total steam consumption. The condenser water quantity must be maintained at full nominal flow for this condition. This also may allow the machine to operate, using very low pressure steam manufactured in a waste heat boiler.

OPERATING COSTS

An important aspect of the economics of an absorption machine other than the first cost is the operating cost.

The annual steam costs may be accurately determined by means of a mechanical integration of steam cost increments for the total operating hours of the machine as follows:

CHART 9—STEAM RATES FOR VARIOUS METHODS OF CONTROL



$$C = c\,(s_1 h_1 + s_2 h_2 + s_3 h_3 + \quad + s_n h_n)$$

where:

C = annual steam costs

$$c = \frac{\text{cost per 1000 lb steam}}{1000}$$

s = steam consumption for incremental percentage of nominal full load (pounds of steam/hour) Steam consumption for each percentage increment is found by multiplying the steam rate (lb/hr/ton) at each percentage increment (*Chart 9*) by the load (tons) at each increment.

h = hours of operation during the year at percentage of nominal full load. (See *Chart 10* for typical graph showing percentage of full load vs. operating hours.)

When comparing differences in operating costs between absorption machines and turbine drive centrifugals, steam rates can be misleading unless the proper steam costs are used for each machine. The amount of heat used or fuel consumed may be identical although the steam rates are considerably different. *Example 1* illustrates that the steam rates for each machine may be different, but the total heat required is the same

Example 1 — Comparison of Heat Requirements

Given:

Chilled water temperature from chiller = 45 F
Available condenser water temperature = 85 F
Absorption machine:
  Steam supply = 12 psig
  Steam rate = 19 lb/hr/ton
Turbine driven centrifugal:
  Steam supply = 125 psig
  Condensing pressure = 26 in. vacuum
  Steam rate = 17 lb/hr/ton

Find:

Amount of heat used for each machine (Btu/hr/ton).

Solution:

Absorption machine
  Total heat of steam at 12 psig = 1161.7 Btu/lb
  Heat of liquid at 212 F leaving machine = 180.0 Btu/lb
  1161.7 – 180.0 = 981.7 Btu/lb steam
  981.7 Btu/lb × 19 lb/hr/ton = 18,700 Btu/hr/ton

Turbine driven centrifugal
  Total heat of steam at 125 psig = 1192.4 Btu/lb
  Heat of liquid at 26 in. vacuum or 125 F = 92.9 Btu/lb
  1192.4 – 92.9 = 1099.5 Btu/lb steam
  1099.5 Btu/lb × 17 lb/hr/ton = 18,700 Btu/hr/ton

This indicates that the amount of heat used for each type of machine is the same even when the steam rates are different.

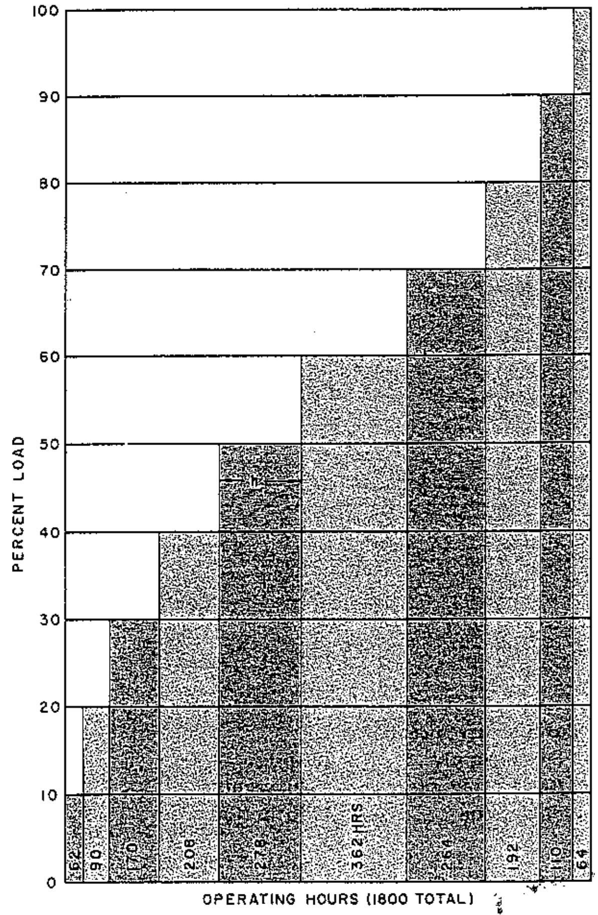
Because a correct analysis of owning and operating costs uses total system heat input as a criterion rather than a straight steam rate per ton of refrigeration, steam costs must be properly calculated and weighted.

The annual power costs of the auxiliary equipment may be calculated as follows:

$$\text{Power costs} = \frac{.746 \times \text{bhp} \times \text{hr} \times \text{cost/kw-hr}}{\text{motor efficiency}}$$

3

CHART 10—TYPICAL GRAPH, PERCENT FULL LOAD VS OPERATING HOURS



STEAM BOILER SELECTION

Any boiler capable of modulating its input to maintain design operating steam pressure within plus or minus one pound is suitable for absorption machine application. This includes:

1 All gas- and oil-fired boilers, since their control is flexible enough to meet this requirement

2 Coal-fired boilers, when the absorption machine never represents more than 15% of the operating load on the boiler. This is because of the slow build-up and shutdown characteristics which limit their flexibility to adjust to the load. Therefore, these boilers are generally limited to large industrial jobs where the steam is being generated in large quantities year-round for other processes.

If the job conditions require that the absorption machine pick up the load rapidly at start-up, it is recommended that the boiler capacity be based on the start-up steam demand of the machine. This demand is the maximum amount of steam that the machine can condense at start-up, and must be obtained from the manufacturer

If the boiler is selected to supply only the full load steam consumption as determined by the machine selection, the boiler temporarily overloads on start-up. This overloading usually affects most boilers by temporarily lowering the steam pressure. This condition is generally not detrimental to low pressure boilers or the absorption machine. If an overload is anticipated, the boiler manufacturer should be consulted for his recommendations.

The net boiler rating should be used to determine its capacity when applied to an absorption machine.

Steam shutoff valves are not necessary for the proper operation of the absorption machine. However, a manual steam shutoff valve is recommended to isolate the machine during long shutdowns

CONDENSATE RETURN SYSTEMS

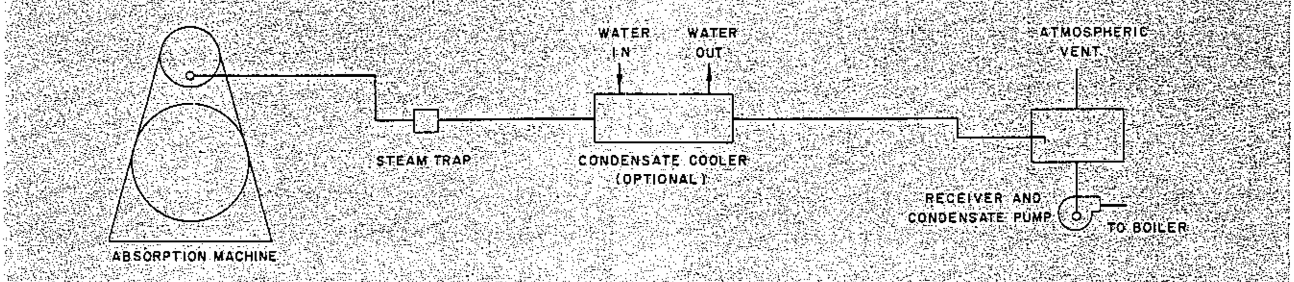
The steam-operated absorption machine requires either a steam trap or a direct return to the boiler thru a wet return arrangement.

If single traps of adequate capacity are not available, multiple traps in parallel should be used. Either an inverted bucket or float and thermostatic trap may be used. The operating steam pressure should be used as the inlet pressure to the trap, neglecting the small pressure loss in the generator tubes. The trap discharge pressure depends on the type of return system, and must be determined for the individual application

A properly sized condensate receiver permits variation of the condensate quantity in the return system from maximum to minimum, with an adequate reserve for the maintenance of boiler feed water requirements

The most common condensate return system is the steam trap vented receiver type (*Fig. 28*). The steam trap insures condensation of all the steam in the absorption machine

A condensate piping arrangement generally known as a wet return (*Fig. 29*) is preferable whenever possible When used under the proper conditions, steam can be returned by gravity from the absorption machine to the boiler without the use of a steam trap. A wet return should not be used if the cost of installation is unreasonably high as compared to the cost of a steam trap discharging into an existing condensate return system. An existing wet return condensate system should be checked for adequate capacity before using.



NOTE: Cooler may be used to heat cold boiler feed water or bleed-off water from cooling tower.

FIG. 28 — SCHEMATIC OF CONDENSATE RETURN USING A STEAM TRAP WITH VENTED RECEIVER

It is generally not practical to utilize an existing vacuum pump condensate return system for an absorption machine because the condensate is far higher in temperature than that for which the return pump was originally selected. This hot condensate flashes and causes vapor binding of the piping and/or vacuum return pump. A separate wet return system is recommended where possible. Where impossible, the condensate may be discharged thru a steam trap to an atmospheric vent receiver and then thru a second trap into the vacuum return system. The condensate may also be cooled to an acceptable level in a heat exchanger and discharged to the vacuum pump condensate system. Any cold water source which may be benefitted by the heat rejected can be used.

CONTROLS

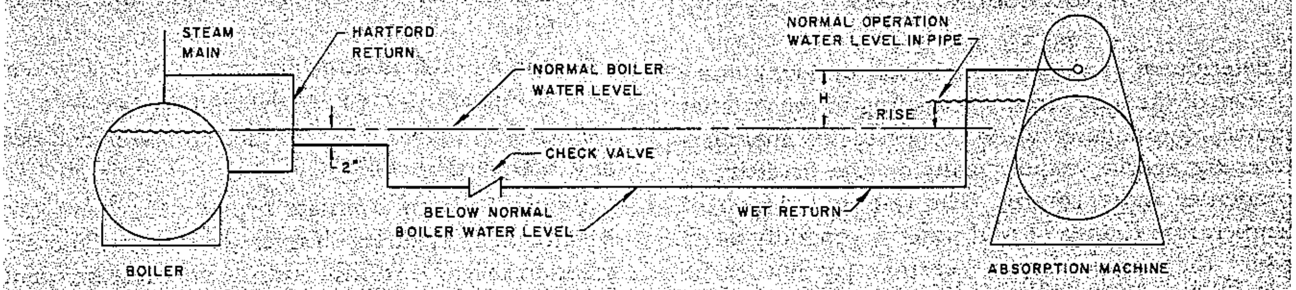
Items which require control are:

1. Condenser water temperature
2. Chilled water temperature
3. Energy source
4. Multiple machines

CONDENSER WATER TEMPERATURE CONTROL

Normally a wide range of condenser water temperatures can be used in the selection of an absorption machine. However, once a particular inlet temperature is established, this must be maintained within definite limits.

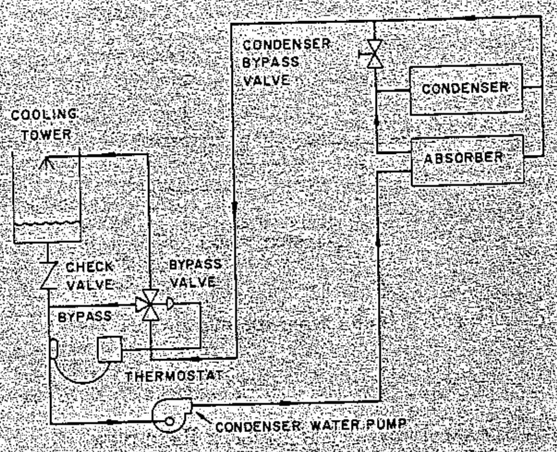
A bypass type control may be required to maintain inlet temperature. The need for bypass control


STEAM MAIN
HARTFORD RETURN
NORMAL BOILER WATER LEVEL
NORMAL OPERATION WATER LEVEL IN PIPE
H
RISE
2"
CHECK VALVE
BELOW NORMAL BOILER WATER LEVEL
WET RETURN
BOILER
ABSORPTION MACHINE

NOTES:

1. Dimension *H* must be greater than the wet return pressure loss plus the difference between boiler and minimum absorption machine steam pressure.
2. *Rise* equals the wet return pressure loss plus the difference between the boiler and the minimum absorption machine steam pressure. Wet return pressure loss is based on a condensate flow equal to the start-up steam demand. Volume represented by *rise* should not be sufficient to flood the boiler when both boiler and machine are shut down.

FIG. 29 — SCHEMATIC OF CONDENSATE PIPING USING A WET RETURN



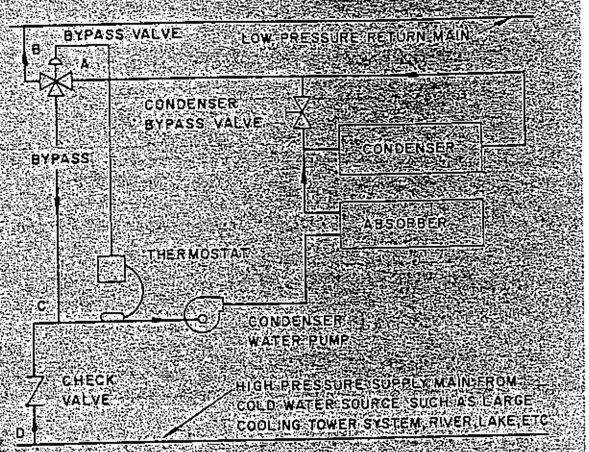
NOTES:

1. Size the bypass 3-way diverting valve and pipe for 100% condenser water flow. Valve pressure loss may be as great as desired.
2. Locate the bypass line and valve close to the cooling tower inlet connection to minimize variations in pump head as the valve position changes.
3. When two absorption machines are installed with a common (or individual) condenser water pump and a common cooling tower, the 3-way diverting valve should be sized for the combined flow of both machines.
4. To keep the system from draining when shut down, shut off the bypass valve when the condenser pump is not operating. Bypass valve port to the cooling tower inlet should be normally closed. Check valve should be installed in the cooling tower drain to prevent water from flowing back into the tower thru the drain.
5. Install the thermostat adjacent to the bypass valve, and the thermal bulb in mixed water adjacent to the bypass line rather than close to the machine.

FIG. 30 — SCHEMATIC OF BYPASS PIPING USED WITH A COOLING TOWER

is determined by the rate and degree of temperature change of the water from the cooling tower or other source of condenser water. Refer to the manufacturer for specific requirements on the necessity of condenser water control. The rate and degree of temperature change of well water is generally negligible; therefore, a bypass control may not be required. The rate and degree of temperature change of water from a cooling tower is generally substantial; therefore, a bypass control is necessary.

The bypass must be capable of limiting the variation in condenser water temperature to 10 degrees and bringing the temperature to operating level quickly. To meet the latter condition, the bypass must always be designed and sized to bypass the total condenser water flow.



NOTES:

1. Size the bypass 3-way diverting valve for 100% condenser water flow.
2. When two absorption machines are installed, the use of separate condenser water pumps and bypass 3-way diverting valves is recommended. Otherwise, size the bypass valve for combined flow to insure a pressure at point *A* that is always higher than the pressure at point *D* plus the pressure differential between points *A* and *C*, even when one machine is valved out of the circuit.
3. Install the thermostat with thermal bulb in the mixed water adjacent to the bypass line rather than close to the machine.

FIG. 31 — SCHEMATIC OF BYPASS PIPING USED WITH A CENTRAL WATER SOURCE

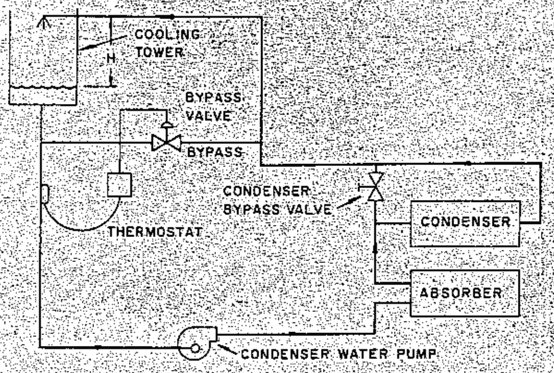
The control system design and valve selection are determined from various combinations of the following:

1. Source of condenser water.
2. Relative locations of the machine and cooling tower.
3. Number of absorption machines and other equipment served by the cooling tower.

Figures 30 and 31 show the most common methods of bypass piping.

Figure 32 is an alternate bypass design and can be used only if the cooling tower is above the absorption machine. Considerable time must be spent to assure that the bypass with the two-way valve is properly applied. Therefore, this approach should only be used when it offers a great economic advantage over the smaller three-way valve.

Figure 33 illustrates throttle control which is applicable to condenser water systems that utilize river, lake or well water when full load capacity is not required as the condenser water temperature drops.



NOTES:

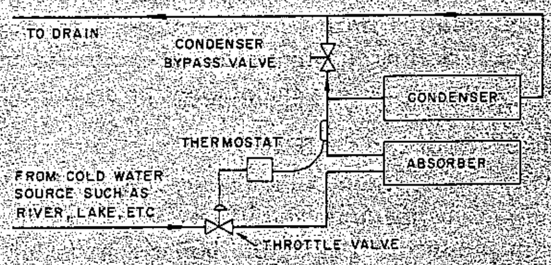
1. Size the bypass valve and pipe for the unbalanced static head of the cooling tower (dimension *H*) and 100% condenser water flow.
2. Locate the bypass line and valve next to the cooling tower base level.
3. If the bypass line and valve cannot be located next to the cooling tower base level, use the arrangement shown in *Fig. 30*.
4. When two absorption machines are installed with a common condenser water pump and a common cooling tower, one bypass line and valve should be installed and sized for the combined flow of both machines and the unbalanced static head of the tower. Locate the valve next to the tower base level.
5. When two absorption machines are installed with individual condenser water pumps and a common cooling tower, individual bypass lines and valves may be installed, each sized and located as specified for a single machine.
6. Install the thermostat adjacent to the bypass valve, and the thermal bulb in mixed water adjacent to the bypass line rather than close to the machine.

FIG. 32 — ALTERNATE SCHEMATIC OF BYPASS PIPING USED ONLY WITH A COOLING TOWER LOCATED ABOVE ABSORPTION MACHINE

These figures illustrate the proper location of the condenser water temperature control valve and schematically show the related condenser water piping. The actual piping layout should be made in accordance with *Part 3, Piping Design*.

When the cooling tower or open drain is below the machine, the piping should contain a loop above the outlet of the condenser nozzle. This prevents the water from draining out of the condenser at shutdown or low flow conditions encountered in throttle type control applications. A vacuum breaker should be installed at the high point of the loop to prevent siphoning of the line.

The bypass valve is usually either a three-way diverting or two-way throttling valve of the globe body type with a linear flow characteristic.



NOTES:

1. Size the throttle valve for 100% condenser water flow.
2. Install the thermostat with the thermal bulb in the condenser water leaving the absorber.

FIG. 33 — THROTTLE CONTROL FOR ONCE-THRU CONDENSER WATER SYSTEMS*

The bypass should be sized and located, and the condenser water pumps selected so that, when water is being bypassed, the flow thru the machine is not increased more than 10%; this prevents overconcentration in the generator and minimizes any increase in pump brake horsepower.

CHILLED WATER TEMPERATURE CONTROL

If an absorption machine is to perform satisfactorily under partial load, a means of effecting a capacity reduction in proportion to the instantaneous load is required. Capacity reduction may be accomplished by steam throttling, control of condenser water flow, or control of reconcentrated solution. For some hot water machines capacity reduction may be accomplished by means of hot water throttling.

These various methods are all used as means to control the ability of the machine to reconcentrate the solution which is returned to the absorber. The more dilute the concentration in the absorber, the less capacity the machine has to chill the water.

Chart 9 shows the comparative performances of these four types of absorption machine capacity control. It is seen that the solution control gives the best steam rate at partial loads; this is where the machine is operated most of the operating season. This lowered steam rate is possible because only enough solution must be reconcentrated to match the load Scaling is minimized because the condensing temperature is maintained at a minimum.

---

*Used if full load capacity is not required when the temperature drops at the cold water source. The ability of the machine to produce full load capacity is not affected when this type of control is applied to condenser water systems where the temperature at the cold water source is constant, as in ground wells. The throttle valve in such systems is used to conserve water rather than maintain condenser water temperature control.

ENERGY SOURCE CONTROL

When using steam as the energy source, the pressure must be maintained within one pound of the design pressure, either by the boiler controls or by a pressure reducing valve if high pressure steam is used.

A back pressure regulator valve to limit steam demand on start-up is rarely used. It may be required if the absorption machine represents most of the load, and if a temporary lowering of the boiler pressure affects operation of the other equipment operated from the boiler. It may also be required where a sudden loss of boiler pressure results in a loss of water in the boiler and/or causes other detrimental effects on the boiler priming.

When high temperature hot water is the energy source, a control valve is usually required to control the hot water flow thru the machine. A two-way throttling or three-way mixing valve is controlled either by a thermostat located in the hot water leaving the machine or by a chilled water thermostat thru a high limit thermostat located in the leaving hot water. The two-way valve should only be used if it does not adversely affect the hot water boiler circulation or the circulating pump. The three-way valve provides a constant flow and is the one most often used

MULTIPLE MACHINE CONTROL

Absorption machines may be applied to parallel and series arrangements of coolers

Installations with machine coolers in parallel may utilize two or more machines With series chilled water flow the cooler pressure loss is cumulative and may become excessive if more than two machines are installed in series.

Parallel Arrangement

When two or more machines are installed with the coolers connected in parallel in the chilled water circuit (*Fig 34*), each machine should control its own leaving chilled water at design temperature as in a single machine installation. The same throttling range should be used for each machine. As the system load is reduced, each machine automatically reduces capacity simultaneously, thus individually producing the same leaving chilled water temperature.

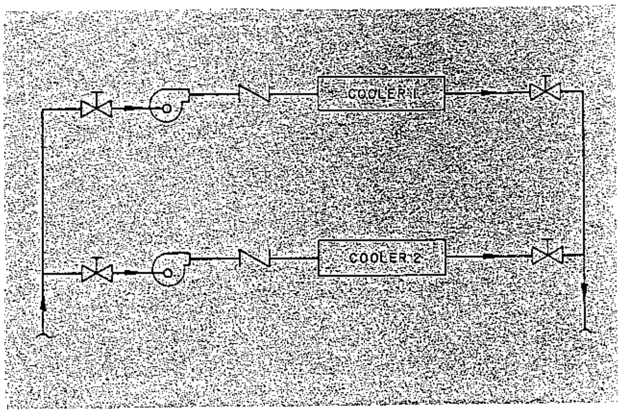


FIG. 34 — PARALLEL ARRANGEMENT OF TWO MACHINES

Operating all machines simultaneously down to minimum load provides the best total steam consumption and the most economical operation when using solution control. There is no economic advantage in shutting down any of the machines at partial load since the steam consumption for two machines operating at partial load is less than for one machine operating at full load. Since there is a minimum of moving parts, there is no reason to shut down a machine to prevent its wearing out.

It is recommended that each machine cooler be provided with a separate chilled water pump on the normal air conditioning application. The pump and pump motors should be selected so that the pump motor is not overloaded if one or more machines and their pumps are shut down

If separate pumps are not provided and a machine is required to be shut down, provision should be made to shut off the chilled water flow and condenser water flow after the shutdown cycle is completed

Series Arrangement

When the coolers are connected in series (*Fig. 35*), equal reduction of loading of each machine produces the best steam consumption. The throttling

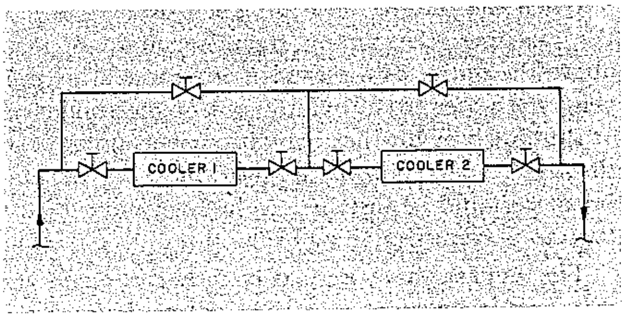
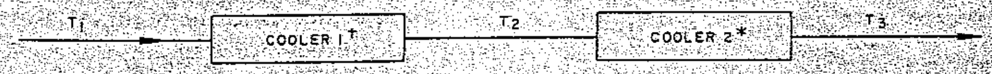


FIG. 35 — SERIES ARRANGEMENT OF TWO MACHINES



System Load (%)	System Cap. (tons)	Cooler 1		Cooler 2		Water Temp (F)			Steam Consumption (lb/hr)		
		Load (%)	Capacity (%)	Load (%)	Capacity (%)	T_1	T_2	T_3	No. 1	No. 2	Total
100	1000	60	100	40	100	60 0	50.4‡	44.0‡	10,830	7,760	18,590
75	750	45	75	30	75	55 25	48.05	43.25	7,790	5,510	13,300
50	500	30	50	20	50	50.5	45.7	42.5	5,200	3,560	8,760
25	250	15	25	10	25	45 75	43.35	41.75	2,610	1,805	4,415
10	100	6	10	4	10	42 9	41.94	41.3	1,095	760	1,855
0	0	0	0	0	0	41.0	41.0‡	41.0‡	—	—	—

*Same size as Cooler 2 in parallel flow.

†Smaller size than Cooler 1 in parallel flow.

‡Difference in temperature from full load to no load equals the throttling range.

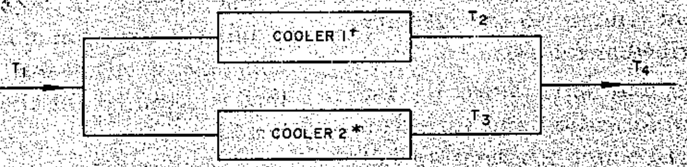
Fig. 36 – Chilled Water Temperature Data For Typical Series-Connected Coolers

range of the high stage machine must be adjusted to insure that each machine handles the same percentage of the system load, both at design and at part load conditions.

In any series selection, the range required on the high stage machine equals the chilled water temperature drop thru the low stage machine plus the throttling range of the low stage machine.

Figure 36 shows the chilled water temperature data from a typical two-machine installation with coolers in series flow. Note that, while the throttling range on the low stage machine (No 2) is the generally recommended 3 degrees, the throttling range on the high stage machine (No. 1) must be adjusted to 9.4 degrees if both machines are to be proportionally reduced to zero load.

Figure 37 shows the chilled water temperature data from a typical two-machine installation with coolers in parallel flow. In this case the throttling range on both machines is identical



System Load (%)	System Cap (tons)	Cooler 1		Cooler 2		Water Temp (F)				Steam Consumption (lb/hr)		
		Load (%)	Capacity (%)	Load (%)	Capacity (%)	T_1	T_2	T_3	T_4	No. 1	No. 2	Total
100	1000	60	100	40	100	60 0	44.0‡	44 0‡	44.0	11,400	7,760	19,160
75	750	45	75	30	75	55.25	43.25	43.25	43.25	8,120	5,400	13,520
50	500	30	50	20	50	50.5	42.5	42.5	42.5	5,300	3,530	8,830
25	250	15	25	10	25	45.75	41.75	41.75	41.75	2,620	1,760	4,380
10	100	6	10	4	10	42 9	41.3	41.3	41.3	1,080	755	1,835
0	0	0	0	0	0	41.0	41.0‡	41.0‡	41.0	—	—	—

*Same size as Cooler 2 in series flow

†Larger size than Cooler 1 in series flow.

‡Difference in temperature from full load to no load equals the throttling range.

Fig. 37 – Chilled Water Temperature Data For Typical Parallel-Connected Coolers

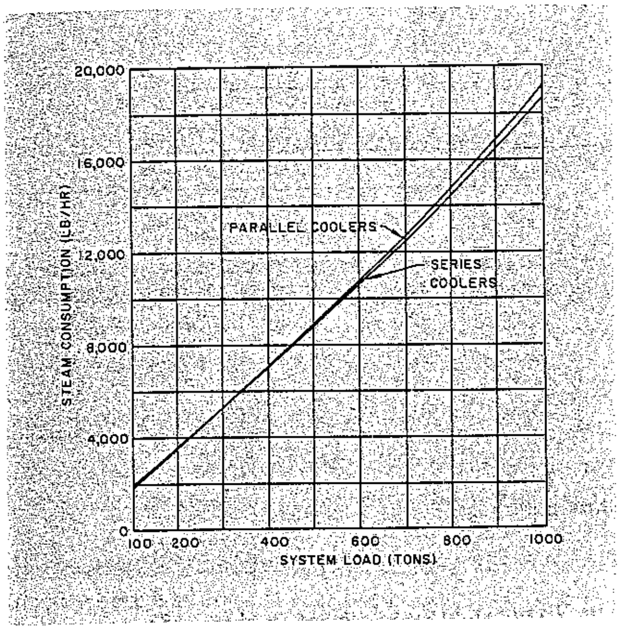


FIG. 38 — PERFORMANCE OF MULTIPLE ABSORPTION MACHINES

Figure 38 indicates the steam consumption for a 1000 ton load for the two types of systems and shows that series operation above about 350 tons uses less steam than parallel operation. The series arrangement has a lower operating cost and generally permits the use of a smaller machine so that the first cost is less.

SAFETY CONTROL

The absorption machine should be provided with safety controls to prevent damage to the machine. These controls are described as follows:

1. Low temperature cutout shuts down the machine to prevent ice formation and tube damage when the chilled water temperature falls below the minimum allowable temperature.
2. Solution pump and evaporator pump auxiliary contacts shut down the machine when either of these pumps become inoperative.
3. Chilled water or condenser water flow switches or their pump's auxiliary contacts shut down the machine when water flow is interrupted in either circuit.

PURGE UNIT

A purge unit is required to remove all noncondensables and to maintain a low pressure in the absorption machine. The purge unit must be able to maintain a pressure below the pressure in the absorber.

INSULATION

The absorption machine requires insulation principally to prevent sweating and the resultant corrosive action on cold surfaces. It may also be used to minimize machine room temperatures and to cover exposed hot lines in or near traffic areas.

Some of the items which may require insulation are:

1. Chilled refrigerant lines and pump
2. Chilled water boxes
3. Generator shell
4. Generator nozzles and headers
5. Solution heat exchanger
6. Hot solution piping.

The cold surfaces may be insulated with flexible fiberglass, closed cell foamed plastic, expanded polystyrenes, plaster or plaster tape, and should include a vapor seal. Water boxes which require removal should be insulated with a removable type of insulation such as sheet metal covers with a granulated fill.

The hot surfaces such as the generator shell may be insulated with a blanket type or low pressure boiler insulation. The generator nozzles and headers should use a removable insulation such as a granulated fill in a sheet metal cover. The hot solution piping requires flexible insulation similar to the types used on the cold piping.

LOCATION

The location of the absorption machine directly influences the economic aspects of the system. A floor adequately strong and reasonably level is all that is required for the location of an absorption machine. However, it is to the engineers advantage to consider other aspects of machine location.

1. It should be located so that the installation costs of the piping between the unit and the equipment it supplies and the wiring and piping of the services to the unit are at a minimum
2. There should be sufficient space near the machine for auxiliary equipment such as chilled water and condenser water pumps and piping.
3. There should be adequate clearances around the machine for access, servicing, and tube pulling or cleaning

Many absorption machines along with their boilers and auxiliaries are installed on upper floors or roofs of buildings because the location has many advantages.

1. It allows the basement areas normally used for such mechanical equipment to be available for profitable use.
2. It eliminates many of the pipes and shafts thruout the building The only services required thru the building are a small fuel line to the boiler, an electric feeder and the normal drain lines
3. Equipment room ventilation is simplified.
4. All mechanical equipment can be located in the same relative area, providing for better maintenance and supervision.
5. It eliminates a long boiler stack and long steam relief lines
6. Pumps and water boxes do not have to be designed for high pressures, as may be expected in tall buildings.

LAYOUT

NOZZLE ARRANGEMENT

The nozzle arrangement chosen for chilled and condenser water should result in a minimum number of chilled and condenser water pipe fittings, should allow proper access to the machine and auxiliary equipment, and should present a neat and attractive appearance

OUTDOOR INSTALLATION

The absorption machine is designed for indoor operation. Outdoor locations are usually not encouraged.

A simple heated structure enclosing the machine is generally preferred, and erection of the enclosure may be less costly than the precautions that may be required. If it is necessary to install the machine outdoors, refer to the manufacturer for his recommodations and precautions.

UNIT ISOLATION

A rubber isolator is normally used under the leg supports of the machine. Such isolation together with the required isolation of the chilled water and condenser water pumps and the piping to and from the machine is usually sufficient for a satisfactory installation.

3

CHAPTER 4. COMBINATION ABSORPTION-CENTRIFUGAL SYSTEM

A combination refrigeration system is well suited to many large tonnage air conditioning systems where operating economy is important and medium or high pressure steam is planned as the driving force.

This chapter includes System Description, System Features, Engineering Procedure and Controls.

As with all applications an owning and operating cost analysis should be made before selecting any specific refrigeration equipment. However, the combination system is generally most attractive on large tonnage applications. These applications include large buildings, big building complexes such as college campuses, and industrial processes requiring water at air conditioning temperature levels

Also, on existing systems where more air conditioning is planned, this increased load can often be added without increasing refrigeration energy requirements by either of the following alternatives:

1. If about ⅓ more load* is planned, a back pressure steam turbine driven centrifugal refrigeration machine can be added to existing absorption equipment
2. If about ⅔ more load* is required, absorption equipment can be added to centrifugal refrigeration equipment. Normally this entails the substitution of a back pressure turbine for a condensing turbine.

As mentioned, medium or high pressure steam must be planned or be available. This can be self-generated or district heating steam

SYSTEM FEATURES

The following are some of the features offered by a combination system:

1. *Minimum Energy Requirements* — Less heat input is required for a combination system than for a condensing turbine-driven centrifugal or an absorption machine alone A correct analysis of owning and operating costs uses total system heat input as a criterion rather than a straight steam rate per ton of refrigeration. *Chapter 3* discusses this point at greater length
2. *Minimum Heat Rejection* — Less heat is rejected from a combination system than from a condensing turbine-driven centrifugal or an absorption machine. Condensation of steam from a back pressure steam turbine in an absorption machine eliminates the need for a more expensive condensing turbine and steam condenser A lower heat rejection to the cooling tower may permit a smaller size tower to be selected

Table 11 compares typical heat input and heat rejection per ton of refrigeration for the three different systems. *Figure 39* is a graphic analysis of *Table 11*.

SYSTEM DESCRIPTION

The following system description and control arrangement chosen for the description is one of many that can be designed for the system. This particular arrangement is presented because it offers control simplicity and a minimum steam rate when operating from full load down to approximately 15-35% load.

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*As explained more fully under *Apportionment of Chilled Water Cooling,* the best operating economy is normally accomplished when the air conditioning load is divided approximately one-third centrifugal refrigeration and two-thirds absorption refrigeration.

TABLE 11—COMPARISON OF STEAM-OPERATED SYSTEMS

	Combination System*	Condensing Turbine Driven System†	Absorption System
Heat Rejection (Btu/ton)	24,600	28,600	30,600
Heat Input (Btu/ton)	12,600	16,600	18,600

*Back pressure turbine 125 psig inlet, 13 psig exhaust, no moisture
†Condensing turbine 125 psig inlet, 4 in. Hg abs exhaust

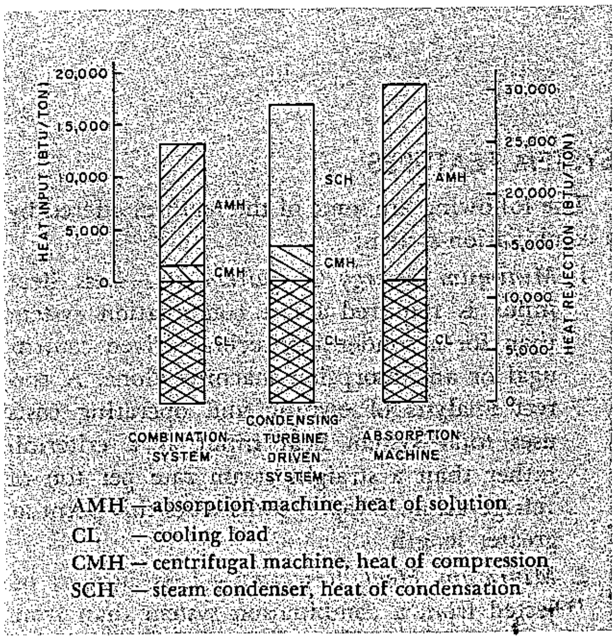


FIG. 39 — GRAPHIC ANALYSIS OF TABLE 11

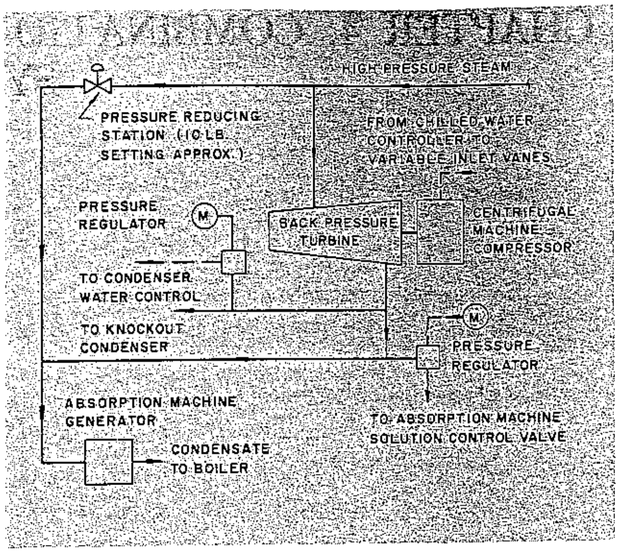

HIGH PRESSURE STEAM
PRESSURE REDUCING STATION (10 LB. SETTING APPROX.)
FROM CHILLED WATER CONTROLLER TO VARIABLE INLET VANES
PRESSURE REGULATOR
M
BACK PRESSURE TURBINE
CENTRIFUGAL MACHINE COMPRESSOR
TO CONDENSER WATER CONTROL
TO KNOCKOUT CONDENSER
M
PRESSURE REGULATOR
ABSORPTION MACHINE GENERATOR
TO ABSORPTION MACHINE SOLUTION CONTROL VALVE
CONDENSATE TO BOILER

FIG. 40 — SCHEMATIC OF COMBINATION SYSTEM STEAM PIPING

Although only one centrifugal machine and one absorption machine are shown in *Fig 40,* a combination of one centrifugal and two or more absorption machines can be used as well. It is shown this way for clarity.

Figures 40 and 41 show the suggested arrangement. Chilled water returns from the load thru the centrifugal machine, to the absorption machine and then back to the load. The condenser water circuit is piped in parallel to the refrigeration equipment with individual pumps for each circuit. This not only allows the versatility of independent machine operation; it also gives good operating economy.

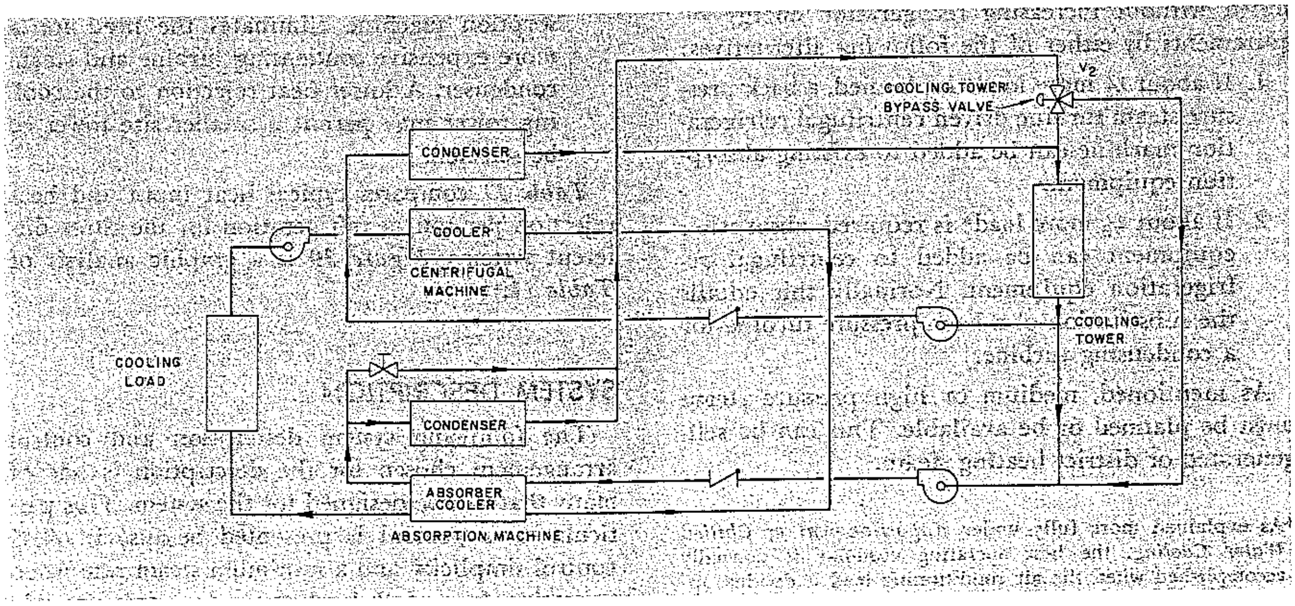


FIG. 41 — *SCHEMATIC OF COMBINATION SYSTEM WATER CIRCUITING*

TABLE 12—MINIMUM SYSTEM STEAM RATE AND CENTRIFUGAL MACHINE PROPORTION OF SYSTEM LOAD

Turbine Inlet Steam Rate (lb/bhp/hr)	Centr. Power Req.* (bhp/ton)	Ratings†	Turbine Exhaust Steam Quality									
			1.00					0.95				
			Absorption Machine Steam Rate at Design (lb/hr/ton)‡									
			17	18	19	20	21	17	18	19	20	21
44	.7	Rate	11.0	11.4	11.8	12.1	12.5	11.3	11.7	12.1	12.5	12.9
		Percent	36	37	38	39	41	37	38	39	41	42
	.8	Rate	11.5	11.9	12.3	12.8	13.2	11.9	12.3	12.8	13.2	13.6
		Percent	33	34	35	36	38	34	35	36	38	39
	.9	Rate	11.9	12.4	12.8	13.3	13.7	12.3	12.8	13.3	13.7	14.2
		Percent	30	31	32	34	35	31	32	34	35	36
	1.0	Rate	12.3	12.8	13.3	13.8	14.2	12.7	13.2	13.8	14.2	14.7
		Percent	28	29	30	31	32	29	30	31	32	33
42	.7	Rate	10.8	11.2	11.5	11.9	12.3	11.1	11.5	11.9	12.3	12.6
		Percent	37	38	39	40	42	38	39	40	42	43
	.8	Rate	11.3	11.7	12.1	12.5	12.9	11.7	12.2	12.5	12.9	13.3
		Percent	34	35	36	37	38	35	36	37	38	40
	.9	Rate	11.7	12.2	12.6	13.1	13.5	12.2	12.6	13.1	13.5	14.0
		Percent	31	32	33	35	36	32	33	35	36	37
	1.0	Rate	12.1	12.6	13.1	13.6	14.0	12.5	13.1	13.6	14.0	14.5
		Percent	29	30	31	32	33	30	31	32	33	35
40	.7	Rate	10.6	11.0	11.3	11.7	12.0	10.9	11.3	11.7	12.0	12.4
		Percent	38	39	40	42	43	39	40	42	43	44
	.8	Rate	11.1	11.5	11.9	12.3	12.7	11.5	11.9	12.3	12.7	13.1
		Percent	35	36	37	38	40	36	37	38	40	41
	.9	Rate	11.5	12.0	12.4	12.9	13.3	12.0	12.4	12.9	13.3	13.7
		Percent	32	33	34	36	37	33	34	36	37	38
	1.0	Rate	11.9	12.4	12.9	13.3	13.8	12.4	12.9	13.3	13.8	14.2
		Percent	30	31	32	33	35	31	32	33	35	36
38	.7	Rate	10.4	10.7	11.1	11.4	11.7	10.7	11.1	11.4	11.7	12.1
		Percent	39	40	42	43	44	40	42	43	44	45
	.8	Rate	10.9	11.3	11.7	12.1	12.4	11.3	11.7	12.1	12.4	12.8
		Percent	36	37	38	40	41	37	38	40	41	42
	.9	Rate	11.4	11.8	12.2	12.6	13.0	11.7	12.1	12.6	13.0	13.4
		Percent	33	35	36	37	38	34	35	37	38	39
	1.0	Rate	11.7	12.2	12.7	13.1	13.5	12.2	12.6	13.1	13.5	14.0
		Percent	31	32	33	34	36	32	33	34	36	37
36	.7	Rate	10.2	10.5	10.8	11.2	11.5	10.5	10.8	11.2	11.5	11.8
		Percent	40	42	43	44	46	42	43	44	46	47
	.8	Rate	10.7	11.1	11.4	11.8	12.1	11.0	11.4	11.8	12.2	12.5
		Percent	37	39	40	41	42	38	40	41	42	43
	.9	Rate	11.2	11.6	12.0	12.4	12.7	11.5	12.0	12.4	12.8	13.1
		Percent	35	36	37	38	39	36	37	38	40	40
	1.0	Rate	11.5	12.0	12.4	12.9	13.3	12.0	12.4	12.9	13.3	13.7
		Percent	32	33	34	36	37	33	34	36	37	38

*A function of the chilled water temperature leaving the centrifugal which is directly affected by the system chilled water rise and leaving temperature (centrifugal on high temperature side).

†Rate is minimum system steam rate in lb/hr/ton. Percent is centrifugal machine proportion of system load.

‡A function of the chilled water temperature leaving the absorption machine (at 100% machine capacity).

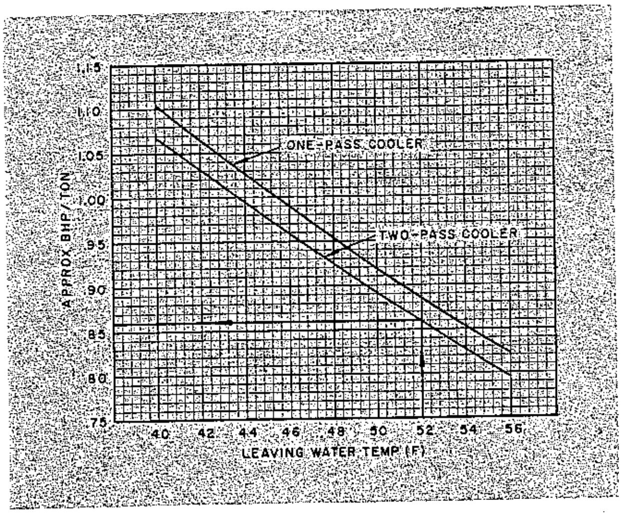
The condenser water temperature is maintained at its required 85 F for the absorption machine, whereas the water entering the centrifugal is allowed to drift downwards at partial loads, thus improving centrifugal operating economies.

The minimum system steam rate occurs when a balanced steam flow condition exists. That is when the absorption machine utilizes exactly all the steam discharged from the turbine. Normal procedure is to accomplish this initial balanced condition at 100% load The control arrangement (described later) accomplishes the balanced steam flow condition at partial loads. *Table 12* shows the minimum system steam rate and the centrifugal portion of the system load for various turbine inlet and absorption machine steam rates.

Control is accomplished as follows. A thermostat in the chilled water circuit leaving the combination system controls the inlet guide vanes or a suction damper in the centrifugal. The absorption machine is controlled by a pressure regulator sensing the steam pressure in the header between the turbine exhaust and the absorption machine. The thermostat maintains a constant chilled water temperature

CHART 11—CENTRIFUGAL COMPRESSOR POWER REQUIREMENTS

Refrigerant 11



and the pressure regulator holds the steam pressure constant ahead of the absorption machine.

Essentially, the chilled water thermostat effects a capacity reduction of the combination system at partial load in proportion to the instantaneous load, while the steam pressure regulator maintains a steam flow balance between the centrifugal and absorption machines and consequently a minimum system steam rate.

ENGINEERING PROCEDURE

Selecting the equipment for a combination system is a matter of achieving the required performance with the minimum owning and operating costs Equipment selection is essentially a trial-and-error process. The following method is recommended to provide a satisfactory system.

PERFORMANCE

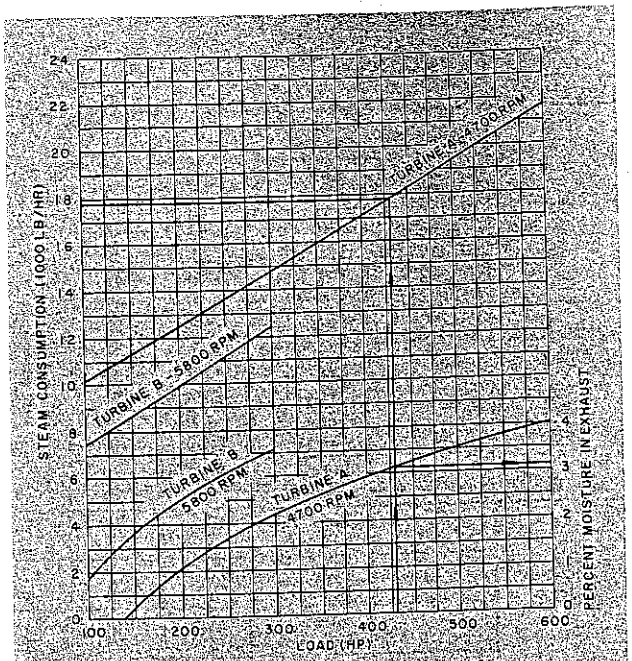
The required performance consists of load or tonnage, chilled water temperature and quantity or rise (F) The load or tonnage is determined by using normal methods. A chilled water rise of 15-20 F is suggested for minimum system steam rate and pressure drop thru the chillers Chilled water temperature selection should not be arbitrary, but should be as high as design permits.

Recommended condenser water quantities are approximately 3 gpm per ton for the centrifugal and 3.5 gpm per ton for the absorption machine.

TURBINE STEAM RATE

Determine the expected single stage turbine steam rate and exhaust steam quality for the specified inlet steam conditions (pressure and superheat) and 13 psig back pressure. (This allows 1 psi pressure loss for the steam piping between the turbine exhaust and the absorption machine inlet, assuming the absorption machine and turbine are close-coupled.)

CHART 12—TYPICAL PERFORMANCE DATA FOR STEAM TURBINES



NOTE: Design steam: 125 psig, 0 superheat, 13 psig exhaust.

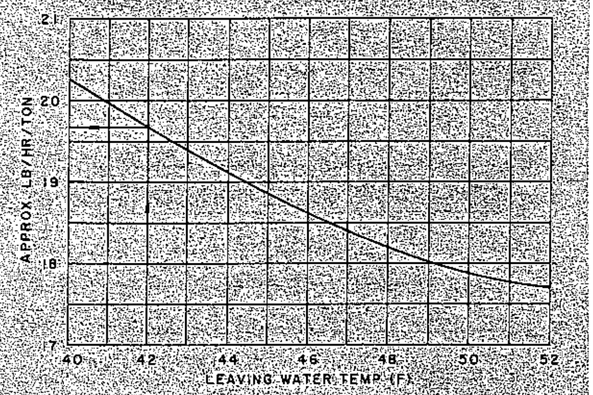
Base the determination of steam rate upon an assumed one-third system load on the centrifugal and a two-thirds system load on the absorption machine. With the centrifugal on the high temperature side a leaving water temperature may be determined From *Chart 11* an approximate brake horsepower per ton of refrigeration may be assumed for the centrifugal.

The required turbine performance is obtained from the turbine supplier and includes the following:

1. Curves relating speed, horsepower and steam consumption from full to minimum load of the centrifugal
2. Curves relating turbine exhaust steam quality to turbine horsepower, speed or steam consumption

There are no turbine performance characteristics which are typical of all turbines. *Chart 12* shows the performance of two typical back pressure turbines for centrifugal refrigeration machine duty.

CHART 13—TYPICAL ABSORPTION MACHINE STEAM RATES



NOTE: 100% machine capacity at all temperatures and 12 psig steam.

DISTRIBUTION OF DESIGN CAPACITY AND MINIMUM SYSTEM STEAM RATE

Calculate the centrifugal machine proportion of the system design load. This is determined from the formula:

$$\text{Load} = \frac{SR_a}{SR_a + (SR_t \times \text{bhp/ton} \times X_{te})} \qquad (1)$$

where:

Load = centrifugal machine proportion of system design load.

SR_a = absorption machine steam rate (lb/hr/ton) at design load.*

SR_t = turbine inlet steam rate (lb/bhp/hr) at design load.

X_{te} = turbine exhaust steam quality.

bhp/ton = centrifugal machine power requirement.†

The system design load minus the centrifugal proportion equals the absorption machine proportion of the load.

Calculate the minimum system steam rate determined from the formula:

$$SR_{min} = \frac{SR_t \times \text{bhp/ton} \times SR_a}{SR_a + (SR_t \times \text{bhp/ton} \times X_{te})} \qquad (2)$$

where:

SR_{min} = minimum system steam rate (lb/hr/ton) of system load.

Example 1 illustrates how these two formulas may be utilized.

Example 1 — Minimum Steam Rate At System Design Load

Given:

2400 gpm of water to be chilled from 57 F to 42 F or a 1500 ton design load.

Centrifugal machine power requirement
= .86 bhp/ton (*Chart 11*) × 500 tons = 430 bhp.

Turbine inlet steam rate at design load

$$= \frac{17{,}700}{430} = 41.2 \text{ lb/bhp/hr } (\textit{Chart 12})$$

Absorption machine steam rate at design load
= 19.65 lb/hr/ton (*Chart 13*)

Turbine exhaust steam quality = .97

Find:

Apportionment of system tonnage

Minimum system steam rate

Solution:

Using formula 1, the centrifugal machine proportion of system design load

$$= \frac{19.65}{19.65 + (41.2 \times .86 \times .97)} = .364$$

then, .364 × 1500 tons = 545 tons

Chilled water temperature range thru the centrifugal
= .364 × 15 F = 5.45 F.

Using formula 2, the minimum system steam rate

$$= \frac{41.2 \times .86 \times 19.65}{19.65 + (41.2 \times .86 \times .97)} = 12.9 \text{ lb/hr/ton}$$

The absorption machine is selected to handle the remainder of the system design load.
1500 − 545 = 955 tons.

APPORTIONMENT OF CHILLED WATER COOLING

Determine the entering and leaving chilled water temperatures for the centrifugal and absorption machine portion of the system design load based on series chilled water flow with the chilled water passing first thru the centrifugal machine. The total required chilled water temperature range is proportioned between the centrifugal and absorption machines in the same ratio as the tonnage. *Example 1* illustrates how the apportionment is determined.

CENTRIFUGAL MACHINE

Select the centrifugal machine for an entering chilled water temperature equivalent to that returning from the load and the leaving chilled water temperature determined previously.

Select the condenser using the available cooling tower water temperature. *Chapter 2* can be used as a guide for the selection of the machine.

ABSORPTION MACHINE

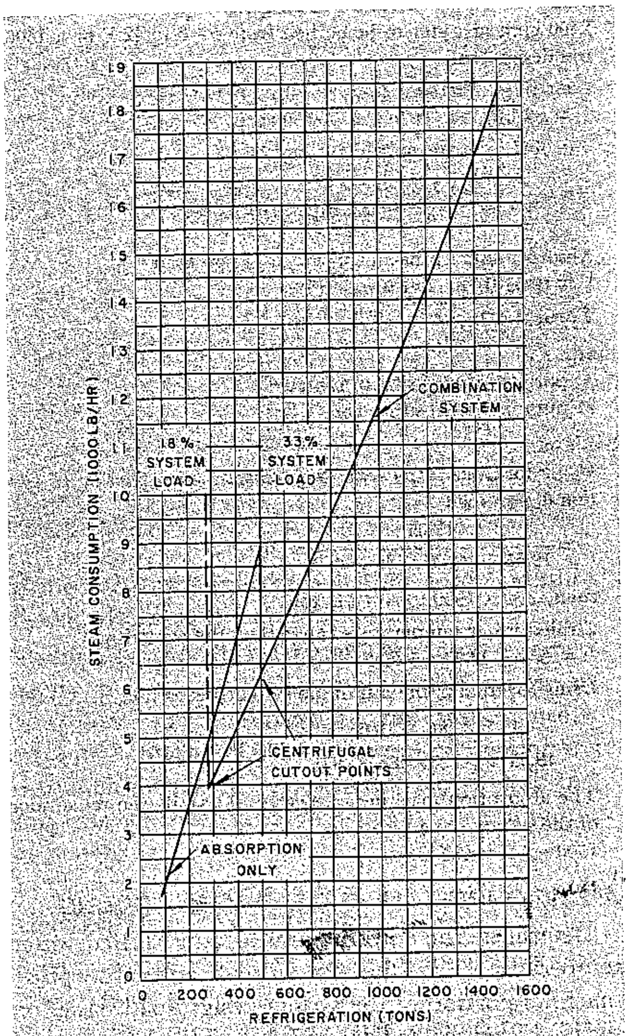
Select the absorption machine for the entering chilled water temperature equal to that leaving the centrifugal and the leaving chilled water temperature required by the design load. Base the machine

---

*For initial determination use approximate lb/hr/ton from *Chart 13* for chilled water temperature leaving the system.

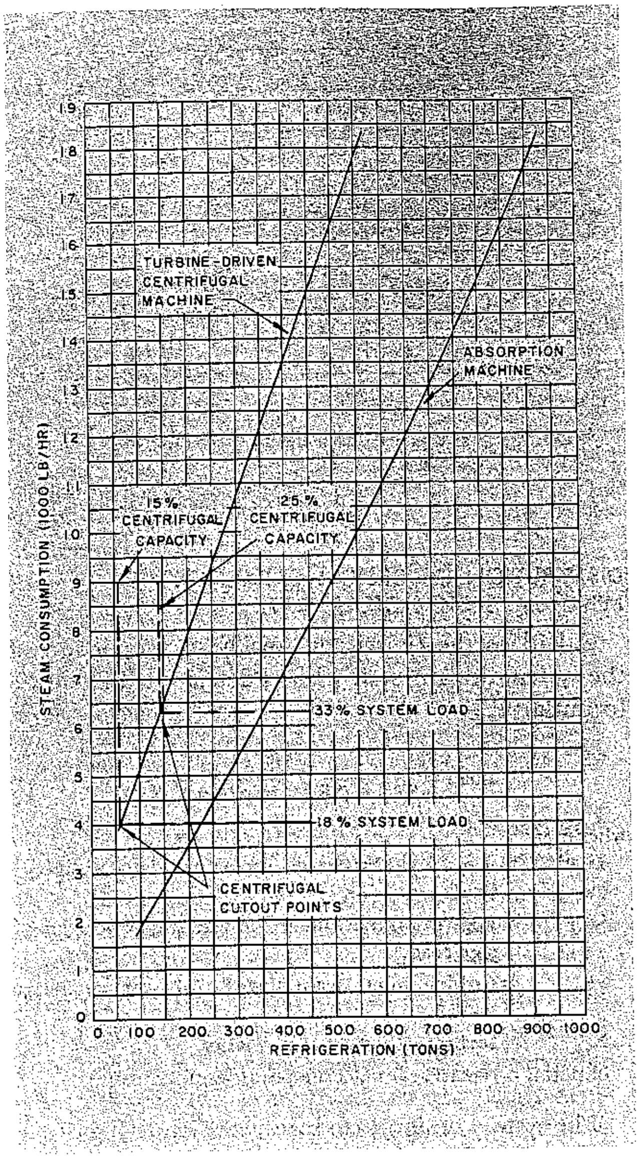
†Use approximate bhp/ton figures from *Chart 11* for chilled water temperature leaving the centrifugal machine.

CHART 14—TYPICAL STEAM CONSUMPTION, COMBINATION SYSTEM



NOTE: Centrifugal machine operates at constant speed within the guide vane control.

CHART 15—TYPICAL STEAM CONSUMPTION, INDIVIDUAL MACHINES



selection on a 12 psig steam supply and the available cooling tower water temperature. *Chapter 3* can be used as a guide for the selection of the absorption machine.

STEAM TURBINE

Select the back pressure steam turbine for the speed and brake horsepower required by the centrifugal at design load. Often a 5% safety factor is added to the required centrifugal brake horsepower before selecting the steam turbine horsepower, to allow for poor entering steam conditions to the turbine.

A single stage turbine is normally used. For equal cost the turbine having the best steam rate should be selected. Two single stage turbines in tandem are recommended in preference to a single, multi-stage turbine. A hydraulic direct-acting governor is recommended for use as the turbine (manual) speed control.

Determine the total required turbine steam flow and exhaust steam quality, and check the balance of steam flow between the turbine and absorption machine. If the quantities determined do not agree with those estimated originally and/or a balance does not exist, further adjustments may be required in the selection of equipment.

Charts 14 and 15 indicate the steam consumption at partial load for a typical combination system and its individual machines respectively.

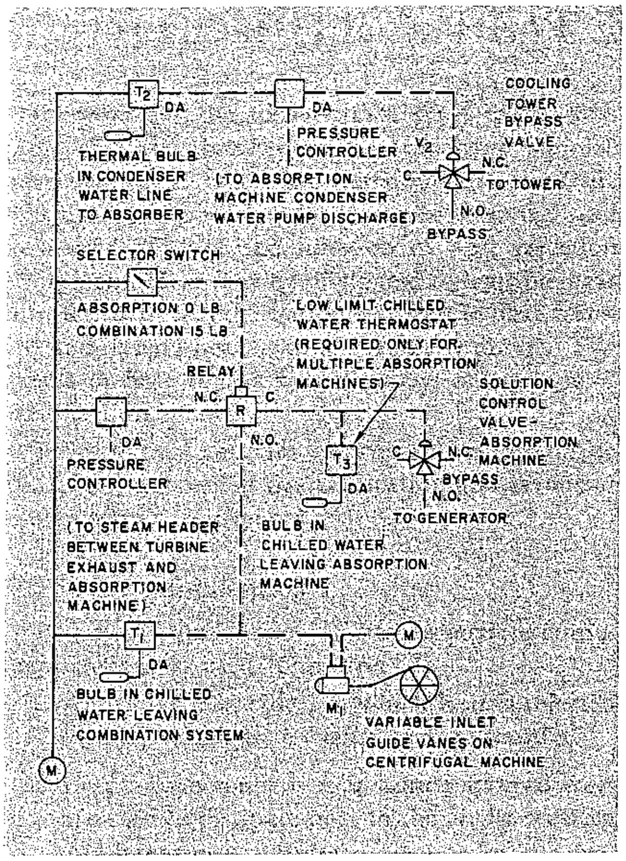

T2
DA
THERMAL BULB IN CONDENSER WATER LINE TO ABSORBER
DA
PRESSURE CONTROLLER
(TO ABSORPTION MACHINE CONDENSER WATER PUMP DISCHARGE)
V2
C
N.C.
N.O.
COOLING TOWER BYPASS VALVE
TO TOWER
BYPASS
SELECTOR SWITCH
ABSORPTION 0 LB
COMBINATION 15 LB
RELAY
N.C.
R
C
N.O.
LOW LIMIT CHILLED WATER THERMOSTAT (REQUIRED ONLY FOR MULTIPLE ABSORPTION MACHINES)
SOLUTION CONTROL VALVE - ABSORPTION MACHINE
T3
DA
C
N.C.
BYPASS
N.O.
TO GENERATOR
DA
PRESSURE CONTROLLER
(TO STEAM HEADER BETWEEN TURBINE EXHAUST AND ABSORPTION MACHINE)
BULB IN CHILLED WATER LEAVING ABSORPTION MACHINE
T1
DA
BULB IN CHILLED WATER LEAVING COMBINATION SYSTEM
M
M
M1
VARIABLE INLET GUIDE VANES ON CENTRIFUGAL MACHINE

FIG. 42 — TYPICAL CONTROL DIAGRAM, COMBINATION SYSTEM

CONTROLS

A typical control diagram for a combination system is shown in *Fig. 42*. Either electrical or pneumatic controls may be used; however, pneumatic controls are most commonly utilized The sequence of operation is the same regardless of which is used.

Capacity control by means of variable inlet guide vanes or a suction damper governs the operation of the centrifugal machine A thermostat with its thermal element in the chilled water line leaving the absorption refrigeration equipment positions the throttling device at the compressor.

A steam pressure regulator controls the operation of the absorption machine. The sensing element of the pressure regulator is piped to the steam header between the steam turbine exhaust and the absorption machine. The regulator positions the solution control valve at the absorption machine in accordance with the steam pressure to control its capacity and maintain a constant pressure. Minimum system steam rate is thus maintained at both full and partial load because the absorption machine utilizes exactly the steam discharged from the turbine.

A selector switch permits operation of the absorption machine alone below the centrifugal cutout point.

COMBINATION OPERATION

With the selector switch in the Combination position the centrifugal machine is under the control of the chilled water controller, and the absorption machine is put under the control of the pressure regulator thru a low limit thermostat (if required).

The cooling tower bypass valve is interconnected with the absorption machine condenser water pump so that the valve is controlled by the condenser water thermostat when the pump is started.

When the temperature falls, the chilled water controller acts to throttle the variable inlet guide vanes at the centrifugal compressor. If the resulting quantity of exhaust steam from the turbine is insufficient to supply the absorption machine in accordance with its solution control valve position, the steam supply pressure drops. The steam pressure regulator senses the pressure drop and throttles the solution control valve to bring the system into balance. An increase in chilled water temperature causes a reverse action.

The low limit chilled water thermostat is a safety device to prevent a freeze-up when more than one absorption machine is required in the design.

ABSORPTION MACHINE OPERATION

When the system load drops below approximately 15-35% of design, the centrifugal machine is shut down. With the selector switch in the Absorption position the chilled water thermostat controls the solution control valve at the machine.

Steam is supplied to the absorption machine thru the pressure reducing station in the bypass line around the turbine. The steam also maintains the turbine hot for easy start-up.

CHAPTER 5. HEAT REJECTION EQUIPMENT

In order for the refrigeration cycle to be complete, the heat absorbed in the evaporator and the heat equivalent of the work required to raise the pressure of the refrigerant must be removed and dissipated. This is the function of heat rejection equipment. Heat may be dissipated by sensible heat transfer or by a combination of sensible heat transfer and latent heat transfer (mass transfer). The means of heat rejection is the basis of equipment classification.

This chapter contains practical information to guide the engineer in the application and layout of heat rejection equipment.

TYPES OF EQUIPMENT

There are three types of heat rejection equipment commonly used. They are:

1. Air-cooled condenser, in which heat is rejected directly to the air by sensible heat transfer.
2. Evaporative condenser, in which sprayed coils are used to dissipate heat to the air by sensible and latent heat transfer.
3. Water-cooled condenser, in which heat is sensibly transferred to water. Although this water may then be wasted, it is usually conserved by a process of sensible and latent cooling in a cooling tower. The water is then recirculated to the condenser. For this reason, the water-cooled condenser and the cooling tower should be examined together as a single heat rejection device.

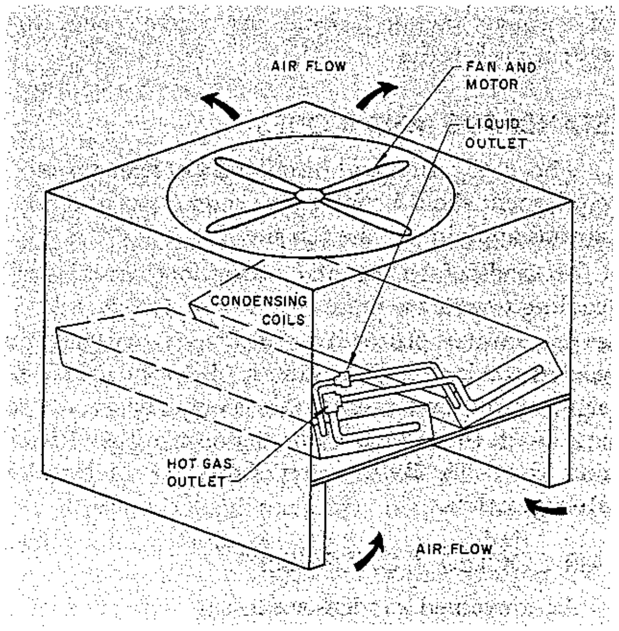


FIG. 43 — AIR-COOLED CONDENSER

Figures 43, 44, 45 and 46 show an air-cooled condenser, an evaporative condenser, a water-cooled condenser, and a cooling tower respectively.

APPLICATION

An evaluation of owning and operating costs is usually the basis for selection of a means of heat rejection. Customer preference and provision for future conditions may influence the choice as well. Local design factors such as air and water conditions and the system application affect the selection insofar as they affect the economics.

In an economic analysis, system size is important since the installed costs per ton of the various condensing methods decrease at different rates with increasing size. All factors being equal, air-cooled condensing is often chosen for capacities up to 75 tons. Evaporative condensing is a primary alternative in the 50-150 ton range. Above 100 tons, water-cooled condensing, in conjunction with a mechanical draft cooling tower, is the most common choice There are many applications where well, river or lake water is used for water-cooled condensing purposes. The installed cost per ton of once-thru water-cooled condensing where the water is wasted remains constant with system size.

In the capacity range where all three condensing methods are alternatives, air-cooled condensing is

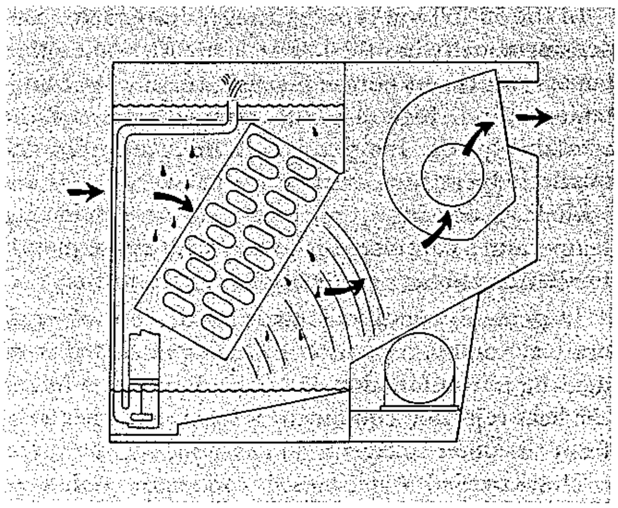


FIG. 44 — EVAPORATIVE CONDENSER

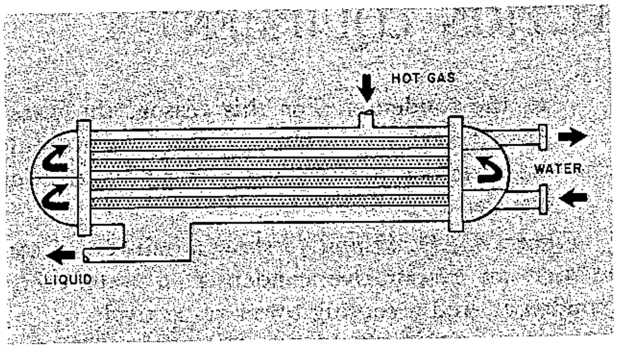


Fig. 45 — Water-Cooled Condenser

usually the highest in first cost. However, maintenance costs for air-cooled condensers are considerably lower for a given capacity. Therefore, air-cooled condensing is well suited to systems where service is infrequent or incomplete. Similarly, long operating hours at light loads favor air-cooled condensing. Over-all operating costs of this method for the commonly applied capacity range are less than for water-cooled condensing, and compare favorably with evaporative condensing.

Other factors supporting the choice of air-cooled condensing are the lack of make-up water or drainage facilities, the availability of only foul water, high summer wet-bulb temperatures, relatively low summer dry-bulb temperatures or high water costs. Installations featuring many independent compressors may possibly be served more satisfactorily by multiple air-cooled condensers or by multiple circuiting in a single air-cooled condenser than by a single evaporative condenser or cooling tower. Also, if operation is required at low outdoor temperatures, air-cooled condensing presents no water freezing problems.

In the 50-150 ton capacity range, evaporative condensing usually has the lowest first cost. Some other factors which encourage its use are low wet-bulb temperatures, high dry-bulb temperatures, or the availability of inexpensive water of adequate quality. Operating costs may be below those of air-cooled condensing, particularly if the condensing temperature considered is lower, with consequently smaller compressor power input requirements.

In general, conditions favoring the use of evaporative condensing also favor water-cooled condensing in combination with a cooling tower. When the heat rejection equipment is located further away from the other refrigeration components, the use of a close-coupled water-cooled condenser and a remote cooling tower becomes economically more attractive. This is because the refrigerant piping necessary with air-cooled or evaporative condensing is more costly than water piping for a given capacity.

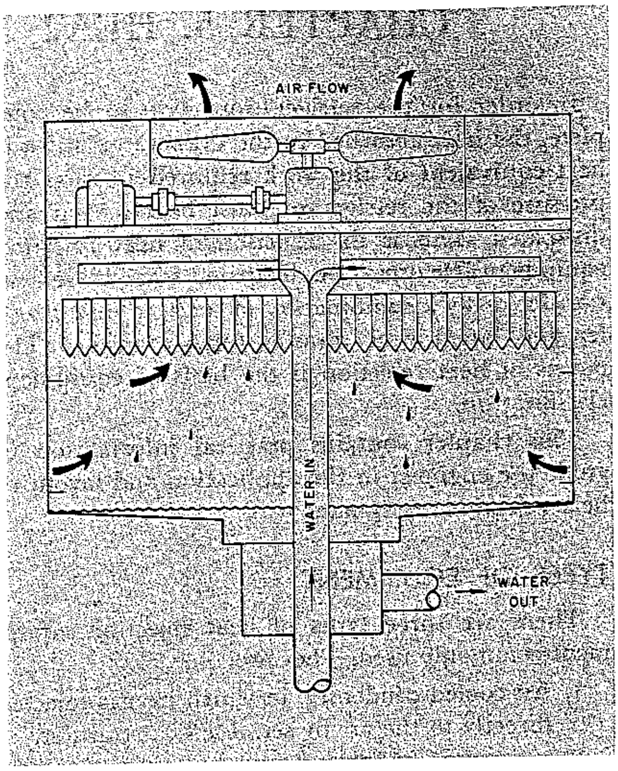


Fig. 46 — Cooling Tower

Once-thru water-cooled condensing may be the most practical and economical choice if there is a nearby supply of water of adequate temperature and quality such as a river, lake or well. Otherwise, city water costs, local codes for the use of water, or lack of adequate sewage facilities may make a once-thru system prohibitive.

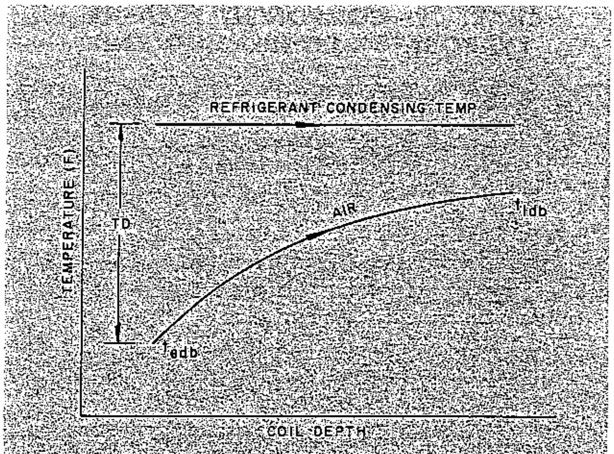
STANDARDS AND CODES

The application and installation of heat rejection equipment should conform to codes, laws and regulations applying at the job site.

Methods of testing and rating mechanical draft cooling towers are prescribed in the ARI Standards, as are similar procedures for evaporative and air-cooled condensers. For water-cooled condensers, design, testing and installations should be in accordance with the ASME Unfired Pressure Vessel Code and the ASA B9.1 Safety Code for Mechanical Refrigeration.

AIR-COOLED CONDENSERS

An air-cooled condenser consists of a coil, casing, fan and motor. It condenses the refrigerant gas by means of a transfer of sensible heat to air passed



t_{edb} = entering dry-bulb temperature
t_{ldb} = leaving dry-bulb temperature
TD = temperature difference

FIG 47 — AIR-COOLED CONDENSING PROCESS

over the coil. The relation between condensing temperature and air temperature is shown in *Fig. 47*

For a given surface and air quantity, the capacity of an air-cooled condenser varies, for practical purposes, in direct proportion to the difference (TD) between the condensing temperature and the entering air dry-bulb temperature. Therefore, assuming the heat rejection requirement is constant, a fall or rise in entering air temperature results in an equal decrease or increase in condensing temperature.

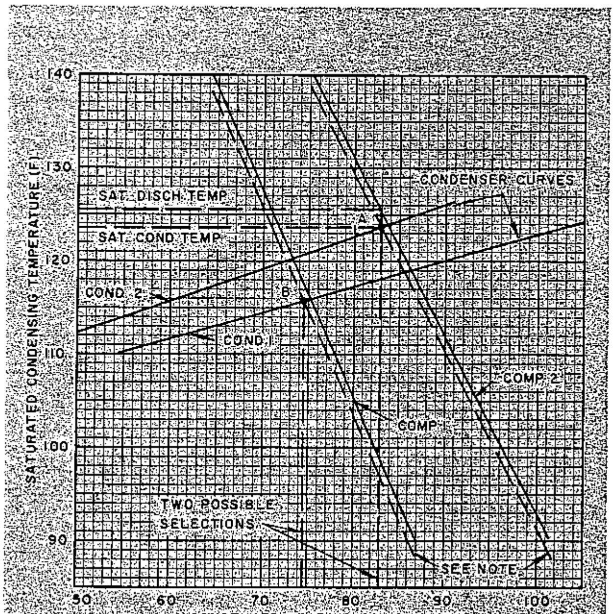
Values for TD range from 15-35 degrees, with condensing temperatures between 110 F and 135 F. In desert areas these temperatures may reach 140 F

UNIT SELECTION

Economics

Air-cooled condensers are most commonly applied to relatively small refrigeration systems. First cost usually dictates the selection of the compressor-condenser combination at conventional condensing temperatures Condenser first cost advantages may be realized with higher condensing temperatures. However, it must be recognized that, as the chosen condensing temperature is increased, compressor power input increases also Higher compressor power requirements may be partially or fully offset by decreased condenser fan motor horsepower. Additionally, common practice is to build most air-cooled condensers with subcooling circuits. This has the effect of increasing total system capacity with a slight increase of compressor power input requirements

CHART 16—COMPONENT COMBINATIONS AIR-COOLED CONDENSING



Refrigerant 12
33 F suction temperature
95 F air temperature entering condenser

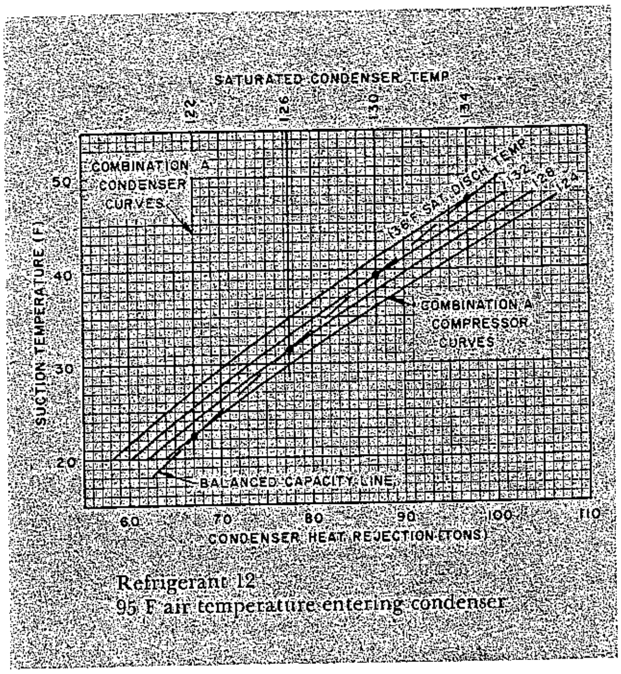
Component Balance

The capacities of a condenser and a compressor operated in combination will balance at some final condensing temperature. Rather than determining this balance point by trial and error, it is preferable to calculate it graphically. This is done by plotting condensing temperature versus heat rejection for both the compressor and the condenser on the same set of coordinates Two or more condensers and compressors can be plotted so that the various combinations can be analyzed for performance, first cost and operating cost.

Chart 16 is a graphical selection for a nominal 60 ton design load. Points *A* and *B* represent two possible combinations Combination *B* has the smaller compressor while combination *A* has the smaller condenser. Compressor power input requirements are higher with combination *A*, but installed first cost is lower

In plotting the air-cooled condenser capacities, it is preferable to use condenser ratings given in terms of total heat rejected rather than in evaporator tons. Compressor heat rejection requirements vary not only with suction and condensing temperature but also with the compressor selected. In addition, heat rejection requirements for hermetic refrigeration machines vary with size. The engineer should refer to specific catalogs for the total heat rejection for

CHART 17—COMPONENT BALANCE, AIR-COOLED CONDENSING UNIT



Refrigerant 12
95 F air temperature entering condenser

each individual compressor for plotting purposes.

The compressor capacity line (shown broken) has been corrected for an assumed 2 degree discharge line loss. Thus, for a given design suction temperature, a point on the corrected line represents a condensing temperature 2 degrees less than the compressor saturated discharge temperature. When designing hot gas lines, it is common practice not to exceed a pressure drop corresponding to a 2 degree change in saturation temperature.

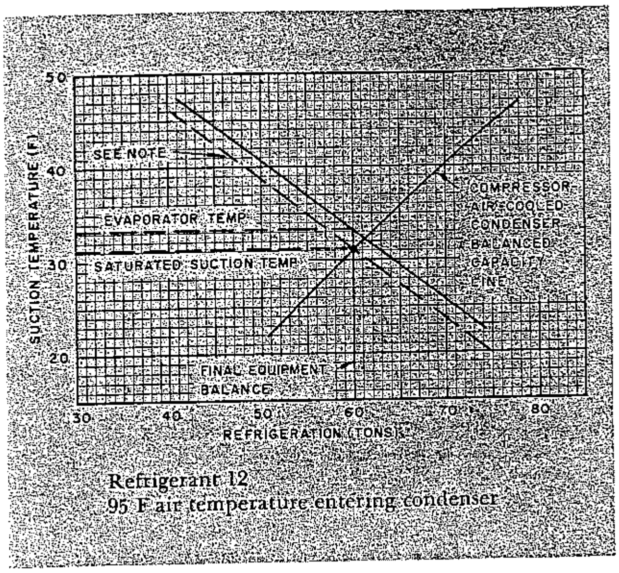
An analysis of combination *A* is given in *Charts 17 and 18*. *Chart 17* describes the heat rejection capacity of compressor-condenser combination A at varying suction and condensing temperatures. The balance line (shown broken) indicates the heat rejection possible with this combination at various suction temperatures, assuming a 2 degree line loss.

Chart 18 includes the balance line shown in *Chart 17* in terms of evaporator tons and, in addition, relates evaporator capacity and suction temperature. The broken line on this chart is the evaporator capacity curve corrected for an assumed 2 degree suction line loss. This standard is not usually exceeded in sizing the suction line.

Compressor Limitations

The selection and application of air-cooled condensers may be limited by the restrictions imposed by the manufacturer on operation of the compressor. These restrictions define maximum compressor saturated suction temperatures and saturated discharge temperatures. Specific data should be obtained from the catalog of the manufacturer involved.

CHART 18—COMPONENT BALANCE, EVAPORATOR-CONDENSING UNIT



Refrigerant 12
95 F air temperature entering condenser

Subcooling

The use of an integral subcooler in an air-cooled condenser provides these operating advantages:

1. An increase in system capacity.
2. A method of offsetting the effects of moderate liquid lifts.
3. Reduced power input per ton of refrigeration.

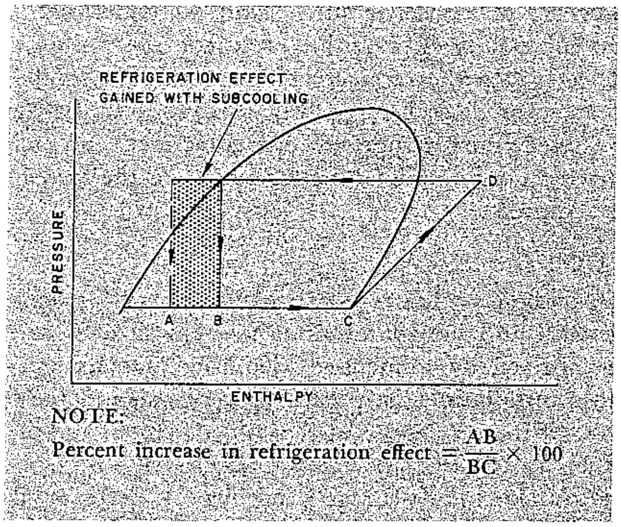
The subcooler is installed in series with the condensing coil, and the condensed liquid from all circuits of the condensing coil is combined before passing thru the subcooler.

Because of the reduced enthalpy of the subcooled liquid, each pound of refrigerant evaporated can absorb more heat, thus increasing the system capacity (*Fig. 48*).

Flashing of liquid refrigerant due to pressure drops from moderate liquid lifts is offset by subcooling.

The power input per ton is reduced because more capacity is available without an increase in the required horsepower. Line *CD* in *Fig. 48* represents the work input which is not changed due to the subcooling.

A receiver (if used) should be valved out of any system using a subcooling coil because the subcooling effect is often offset by the liquid flashing in the receiver.



NOTE:
Percent increase in refrigeration effect $= \frac{AB}{BC} \times 100$

FIG. 48 — EFFECT OF LIQUID SUBCOOLING

Atmospheric Corrections

Air-cooled condenser ratings are based on air at standard atmospheric conditions of 70 F and 29.92 in Hg barometric pressure. If a condenser is equipped with a direct drive and is to operate at an altitude above sea level, heat rejection ratings should be corrected for the change in air density. This correction amounts to a decrease in capacity of approximately 9% at an altitude of 5,000 feet. However, it should be pointed out that this reflects only about a 3% loss in total system capacity.

On condensers equipped with belt-driven equipment, fan speed can be increased to offset this correction.

Capacity corrections for air temperature deviations are unnecessary unless temperatures exceed 125 F.

Multiple Circuiting

Some installations may include several independent refrigeration systems, each operating at the same or different suction or condensing temperatures. These may be supplied with a single condenser which has a multiple of circuits, each operating with a separate refrigeration system.

CONDENSER CONTROL

There are two basic methods of controlling the capacity of air-cooled condensers. They are air side and refrigerant side control. The air side controls utilize methods of varying the air flow thru the condenser. The refrigerant side controls vary the amount of available condensing surface by flooding portions of the condenser with liquid refrigerant.

Air-cooled condenser controls are needed to maintain a sufficient pressure differential across the refrigerant expansion device to provide the required refrigerant flow to offset the load. The control should operate satisfactorily at any outdoor temperature at which refrigeration is required. The minimum outdoor temperature above which the control system operates satisfactorily must be obtained from the manufacturer of the control. This minimum temperature may depend on whether the refrigeration system can operate unloaded some of the time or if it operates loaded all of the time. The application of the refrigeration system, whether for liquid chilling, direct expansion cooling or a heat pump, may also influence the minimum temperature.

Most manufacturers of air-cooled condensers offer patented methods of control which can only be used with their condensers. Refer to these manufacturers for their recommendations concerning operating limits and applications for their controls.

Refrigerant migration to the condenser on shutdown may be prevented or its effect on a proper start-up may be lessened by the condenser control. Refrigerant migration occurs because the condenser is in a colder atmosphere than the other parts of the system. This may create trouble on start-up since the condenser may be full, or partially full, of subcooled liquid which is at or close to the outdoor temperature. There may be very little pressure difference across the expansion valve and the system may cycle on the low pressure switch when attempting to start.

Several methods are available to aid in providing for winter start-up. One method is to shunt out the low pressure switch at start up with a time delay relay until the system gets up to pressure. Another method is to locate the low pressure switch in the liquid line and put a defrost thermostat on the coil. When single pump out control is used, a low pressure stat is also required in the suction line.

Feeding subcooled liquid to a liquid chiller may cause frost pinching of the tubes or a freeze-up of the chiller due to erratic operation of the expansion valve. With a direct expansion cooling coil, the subcooled liquid may not allow proper distribution of the refrigerant thru the parallel circuits in the evaporator. Frost may deposit on the coil and can build up and block off the air flow.

Proper selection of the expansion valve and the selection of an adequate receiver to hold any excess refrigerant may also be required in connection with the condenser control.

Air Side Control

Examples of the air side controls include the cycling of fans in sequence when multiple fans are used on a single coil system; the modulation of a volume damper installed in the fan discharge or as a face damper on the coil; the modulation of a bypass damper installed to bypass air around the coil; a variable speed fan; or possibly a combination of these controls.

Refrigerant Side Control

Types of refrigerant side controls include an electrically heated surge type receiver to maintain a minimum liquid temperature in the receiver; a bypass valve to bypass hot discharge gas around the condenser to maintain a minimum downstream or receiver pressure; a pressure regulating valve in the drain line from the condenser to maintain a minimum pressure in the condenser; or variations and combinations of these controls.

LOCATION

An air-cooled condenser may be located indoors or outdoors. It may be remote or near, above or below the compressor. The greater the distance separating the condenser and compressor, the greater is the first cost and operating cost. Specific location recommendations include the following:

1. Locate the unit so that air circulates freely and rapidly without recirculating.
2. Locate the unit away from areas continuously exposed to loose dirt and foreign matter.
3. Locate the unit away from occupied spaces with low ambient sound levels.

Condenser location with respect to the evaporator can have an influence on the liquid line size. Since most air-cooled condensers are manufactured with a coil designed to give liquid subcooling, the liquid line can be designed for a much higher pressure drop when the evaporator is below the condenser than when the evaporator is located at the side of or above the condenser. It is suggested that this subcooling effect and pipe sizing be studied because of the increased cost of pipe installation and additional refrigerant required for an oversized liquid line.

It may be possible to utilize building exhaust air as part or all of the supply to the condenser, thus reducing the size of the condenser or lowering the head pressure.

When air-cooled condensers are manufactured with the subcooling coil integral with the condensing coil, liquid receivers are not normally used. The use of a receiver often completely eliminates the subcooling effect if the liquid condenser drains directly into the receiver before passing on to the evaporator. Receivers (when used) are normally bypassed during normal operation and are restricted for pumpdown when service on the system is required.

LAYOUT

Air-cooled condensers are manufactured for both vertical and horizontal air flow. Vertical coils can be affected by condenser orientation. On an outdoor installation the prevailing winds should blow toward the air intake of the unit. If this is impossible, a discharge air shield is suggested. Also, special intake and discharge hoods may be necessary if snow can accumulate in the fan section. Orientation has no effect on the performance of air-cooled condensers equipped with horizontal coils.

For indoor installations, fresh air intakes and discharge ducts to the outdoors must be provided. It is necessary to arrange these ducts so that recirculation does not occur.

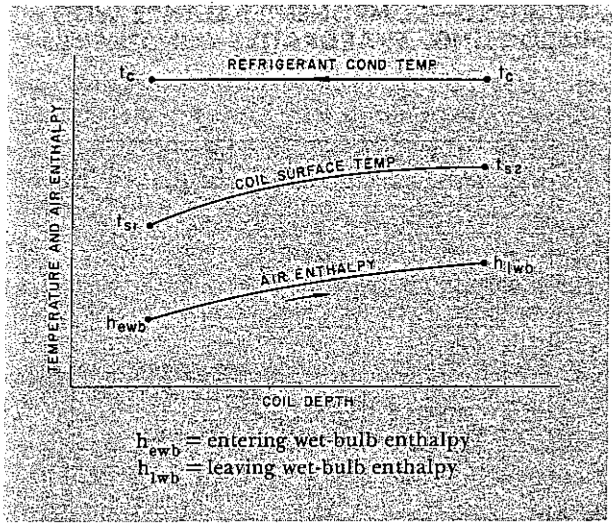
Refer to *Part 3, Chapter 3* for refrigerant piping details.

EVAPORATIVE CONDENSERS

An evaporative condenser consists of a condensing coil, fan and motor, water distribution system, sump, recirculating pump, and casing. It condenses the refrigerant gas by means of a combination sensible and latent heat transfer process. Rejected heat is dissipated by water diffused over the coil surface. It is then transferred to the air passing over the coil. Latent heat transfer is more effective as a means of heat dissipation and, therefore, permits a unit with less cubage than an equivalent air-cooled condenser. The relation between condensing temperature, air enthalpy, and coil surface temperature is shown in *Fig. 49.*

The capacity of an evaporative condenser may be increased either by lowering the entering air wet-bulb temperature or by increasing the condensing temperature. Condensing temperatures normally range from 100-115 F. Increasing the air quantity beyond the design has little effect on capacity.

An evaporative condenser may be used to cool other liquids such as oil and water instead of a refrigerant. Most manufacturers provide ratings for such applications. For optimum performance, piping must be designed so that the flow of water thru the condenser coil is opposite in direction to the flow of air.



h_{ewb} = entering wet-bulb enthalpy
h_{lwb} = leaving wet-bulb enthalpy

FIG. 49 – EVAPORATIVE CONDENSING PROCESS

UNIT SELECTION

The normal design wet-bulb temperature of a locale should be the basis of an evaporative condenser selection since higher wet-bulb temperatures seldom occur, and then only briefly. If design conditions must be maintained at all times, such as in some industrial process applications, the maximum wet-bulb temperature should be used to insure adequate equipment capacity.

Economics

As in the case of air-cooled condensers, first cost usually dictates the selection of an evaporative condenser-compressor combination. Operating at relatively high condensing temperatures with a heavily loaded condenser lowers the installed cost per ton but may increase the operating cost per ton. For over-all maximum economy of operation, a difference of 25-30 degrees between condensing and entering air wet-bulb temperatures is suggested.

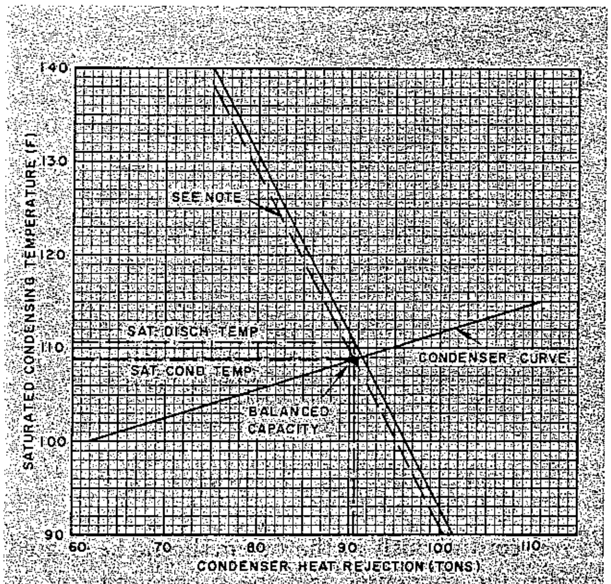
Component Balance

The balanced capacity of an evaporative condenser-compressor combination may be determined graphically as described under *Air-Cooled Condensers*. *Chart 19* illustrates such a graphical analysis at a given wet-bulb temperature and air quantity. As mentioned previously, it is preferable to base the condenser operating characteristic on a known condenser heat rejection performance rather than on condenser ratings in tons of refrigeration effect.

Subcooling

The use of an integral or accessory subcooling coil with an evaporative condenser provides the following operating advantages:

CHART 19—COMPONENT BALANCE, EVAPORATIVE CONDENSING



Refrigerant 12
33 F suction temperature
76 F air wet-bulb temperature entering condenser

1. An increase in system capacity.
2. An offset to the effect of moderate liquid lifts.
3. Reduced power input per ton of refrigeration.

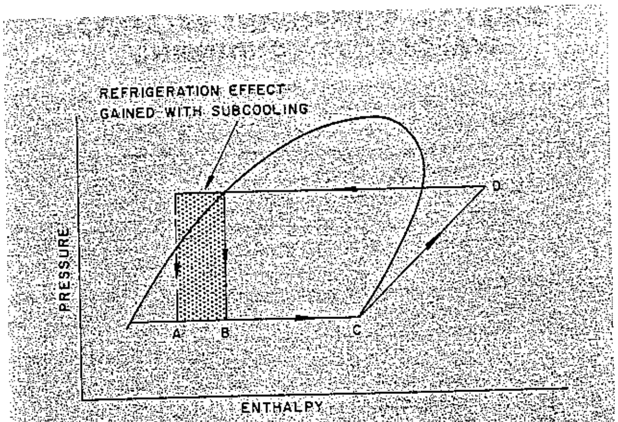
The subcooling coil is installed in series with the condensing coil and is the first coil contacted by the entering air.

When the balanced capacity of a condenser-compressor combination is slightly below design requirements, it is usually more economical to add a subcooling coil than to select the next larger combination.

Figure 50 is a pressure-enthalpy diagram illustrating the source of the additional capacity realized from the use of subcooling. The capacity of the condenser-compressor combination is increased because with subcooled liquid each pound of refrigerant evaporated does more work. *Chart 20* shows the effect of a subcooling coil on the same combination as is shown in *Chart 19*.

The amount of increase in system refrigeration capacity as a result of subcooling is obtained from the manufacturers' evaporative condenser ratings.

Liquid subcooling cannot be obtained if a receiver follows the subcooling coil in the refrigeration system. If a receiver is used and if it must be so located, it should be used as a storage vessel only and valved out of the circuit during operation. A



NOTE:
Percent increase in refrigeration effect $= \frac{AB}{BC} \times 100$

Fig 50 — Effect of Liquid Subcooling

receiver continuously in the system should be located between the condenser and the subcooling coil in order to obtain the subcooling effect.

Atmospheric Corrections

At elevations above sea level, the reduction in weight flow of air due to the decrease in density of the air at high altitude is offset by the greater latent heat absorbing capacity of the high altitude air.

Fan motor selections made on a sea level basis are conservative.

To determine actual motor brake horsepower at the design fan speed at any elevation, multiply the power requirement at sea level by the ratio of the air density at the design altitude to the air density at sea level.

Multiple Circuiting

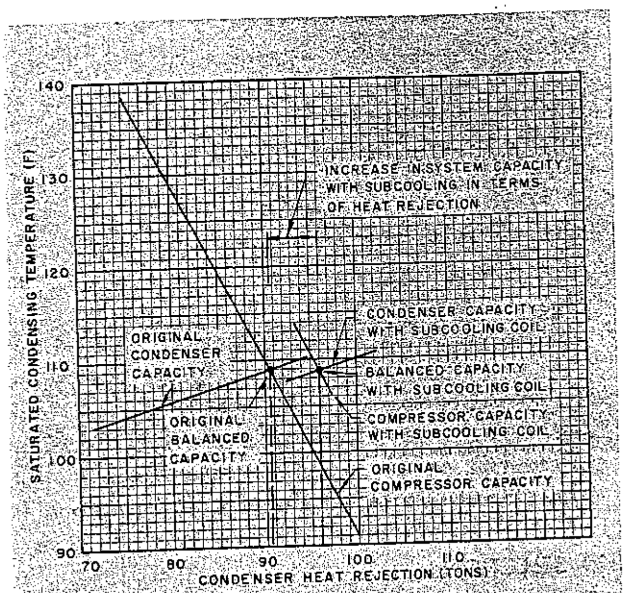
For applications in which one condenser serves several separate refrigeration systems, evaporative condensers are available with headers divided to form two or more independent refrigerant circuits. The number and relative capacities of these individual circuits depend on the circuit design of the condenser coil. Individual circuits may be operated at the same or different suction or condensing temperatures.

CONDENSER CONTROL

Operation of an evaporative condenser at low outdoor temperatures requires special consideration in order to prevent the freezing of recirculation water and to insure proper functioning of the thermostatic expansion valve.

The following methods may be used to maintain condensing pressure in an evaporative condenser:

CHART 20—COMPONENT BALANCE WITH SUBCOOLING, EVAPORATIVE CONDENSING



Refrigerant 12
33 F suction temperature
76 F air wet-bulb temperature entering condenser

1. An automatic discharge damper which varies the flow of air across the coil.
2. An automatic air recirculating assembly (*Fig. 51*) which controls the entering air wet-bulb temperature.
3. A fan motor having two or more speeds to vary the air flow across the coil.

Automatic discharge dampers or air recirculation damper assemblies are the most satisfactory solutions

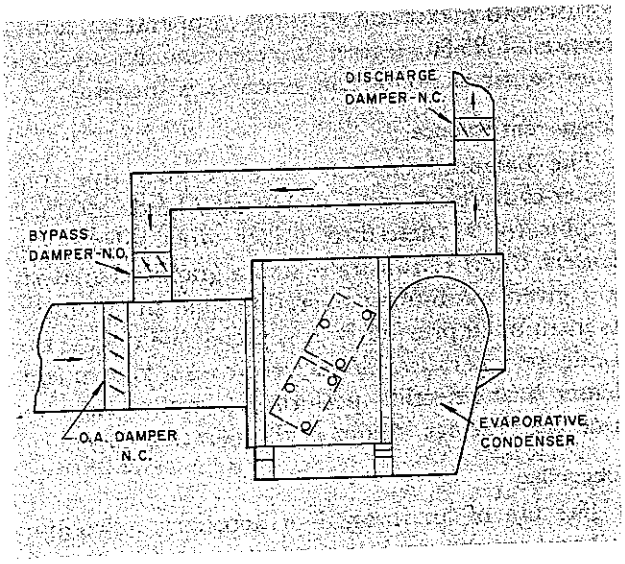


Fig 51 — Air Recirculating Assembly

to the problem of positive condensing pressure control. These dampers are often sold as accessories by the condenser manufacturer. The dampers may be actuated in response to a condensing pressurestat or, in the case of recirculation control, in response to a recirculation water thermostat.

Condensing pressure control methods involving a cycling of the recirculation pump are not recommended; cycling of the pump produces a rapid accumulation of scale deposits

WINTER OPERATION

When year-round operation is required, it is advisable to locate the unit indoors but, if an outdoor installation is needed, the unit should be operated dry with the sprays shut off and the water drained from the pan. This generally reduces the capacity of the unit, but full capacity is usually not required at this time.

LOCATION

An evaporative condenser may be located indoors or outdoors. An indoor location requires ductwork which increases first cost and fan power requirements; it is recommended if year-round operation is planned; it minimizes problems of unreliable start-up and the freezing of recirculated water. An indoor condenser operated in the summer only should be provided with manual intake and discharge dampers in the duct in order to prevent the introduction of cold air and attendant problems of moisture condensation.

A condenser mounted outdoors usually requires no protective covering, but should be provided with drainage facilities for the tank, pump and water piping.

Other location recommendations are listed under *Location* for air-cooled condensers

LAYOUT

Whether indoors or outdoors, the unit should be installed elevated above the floor, roof or ground. This may be accomplished by suspending the unit or by providing a mounting pad.

MAKE-UP WATER

Provision must be made for city water make-up for evaporation and bleed-off water losses. Evaporation occurs at the rate of approximately 1.8 gallons per hour per ton of refrigeration. Bleed-off varies from 50-200% of evaporation, depending on water treatment recommendations. Refer to *Part 5, Water Conditioning*.

Refer to *Part 3* for refrigerant piping details.

WATER-COOLED CONDENSERS

A water-cooled condenser consists of heat transfer tubes mounted within a steel shell. Condenser water passes thru the tubes, and the condensing refrigerant occupies the shell surrounding the tubes. The shell is equipped with a hot gas inlet, a liquid sump, purge connections, water regulating valve connection, and a pressure relief device. The shell-and-tube condenser has a supporting tube sheet at each end and removable heads while the shell-and-coil condenser has a spiral or trombone tube bundle, accessible from one end only. Tubes must be cleaned chemically instead of by reaming or brushing. Shell-and-coil condensers are of relatively low cost and are often used in air conditioning applications.

At a given entering water temperature and condensing temperature, condenser capacity is dedecreased by decreasing the water flow rate, thus increasing the water temperature rise. Condensing temperatures usually range from 100-110 F, but may be as low as 80 F for once-thru city water condensing.

UNIT SELECTION

The water-cooled condenser and the cooling tower should be considered as a single heat rejection device for the purposes of selection and application. Therefore, the economics of condenser selection are reviewed under the subject of *Cooling Towers*.

In a once-thru application, the entering water temperature used for selection should be the maximum water temperature prevailing at the time of maximum refrigeration load.

Selection of the number of water passes should be made with regard to the temperature and pressure of the water available. The low pressure and higher temperature water available from cooling towers generally dictates the least number of passes.

Normally, manufacturers base water-cooled condenser ratings on various conditions of the tube scaling on the water side. A fouling factor represents the resistance to heat flow presented by scaling. Since some surface fouling is present on tube surfaces from the beginning of operation, a minimum fouling factor of .0005 is suggested for selection. Scale factors for various types of condensing water systems are shown in *Part 5*.

The factors shown should be tempered by operating conditions. A reduction in the factor is justified in the case of frequent cleaning, an unusually low condensing temperature, or when operation is less than 4000 hours annually.

CONDENSER CONTROL

Control of water flow thru condensers may be required to limit the condensing pressure to a predetermined minimum. Two methods of restricting this flow are commonly used, a two-way throttling valve or a three-way diverting valve.

The two-way valve is useful in maintaining condensing pressure in once-thru applications utilizing city, well, lake or river water. In the case of city water, a prime objective may be to minimize water costs.

The three-way valve is most often used with a cooling tower. It operates to direct water around the condenser as the condensing temperature is lowered. This allows the pump to maintain its flow and reduces problems of water distribution with multiple unit applications.

COOLING TOWERS

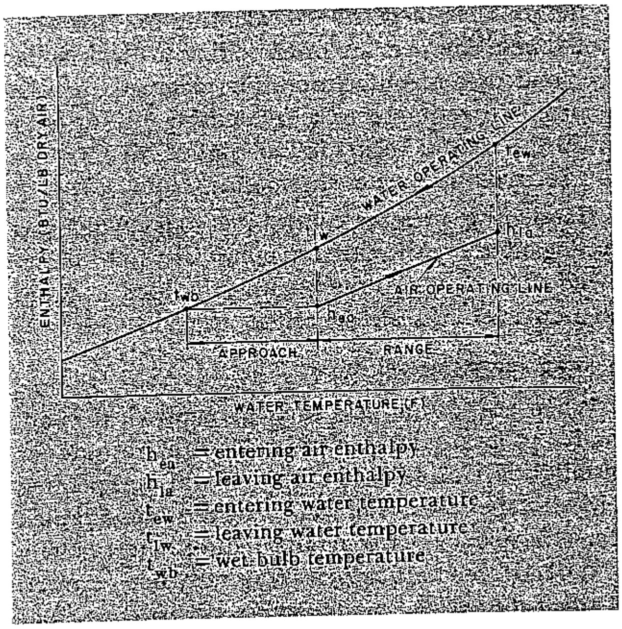
Atmospheric water cooling equipment includes spray ponds, spray-filled atmospheric towers, natural draft atmospheric towers, and mechanical draft towers. Except for relatively small installations on which the spray-filled atmospheric tower may be used, the mechanical draft tower is the most widely used for air conditioning application. Of the types of equipment available, the mechanical draft tower is the most compact, the lowest in silhouette, the lightest and the best suited to meet exacting conditions of water temperature.

Air flow thru a mechanical draft tower may be forced or induced. Referring to the direction of air flow relative to the water flow thru the fill, a tower may be classified as counter-flow, cross-flow or parallel flow. The towers commonly used are induced draft, counter-flow or cross-flow.

A cooling tower consists of a casing, basin and sump, water distribution system, fill, fan, motor and drive.

The relation between the enthalpy of the air and the water temperature is illustrated in *Fig. 52* for a counter-flow tower. The rate of heat transfer from the water to the air depends on the enthalpy of the air which is represented by wet-bulb temperature. This rate is independent of the air dry-bulb temperature. For a given air and water quantity thru a tower, the rate of heat transfer, or rated tower capacity, is increased by lowering the entering air wet-bulb temperature requirement or by raising the temperature of the water entering the tower.

Tower performance is specified in terms of water range and approach. Cooling range is the difference between water entering and leaving temperatures and is equal to the temperature rise thru the condenser. Approach is the difference between the water temperature leaving the tower and the entering air wet-bulb temperature.



h_{ea} = entering air enthalpy
h_{la} = leaving air enthalpy
t_{ew} = entering water temperature
t_{lw} = leaving water temperature
t_{wb} = wet-bulb temperature

FIG. 52 – WATER COOLING PROCESS, COUNTERFLOW COOLING TOWER

UNIT SELECTION

A cooling tower should be selected for the normal design wet-bulb temperature of the locale. If design conditions must be maintained at all times such as in some industrial process applications, the maximum wet-bulb temperature should be used to insure adequate equipment capacity.

Economics

The selection of refrigeration equipment and a cooling tower is influenced primarily by the condensing temperature chosen and its effect on tower range and approach. Relatively small increases in condensing temperature can produce large economies in tower size and cost, weight, space required and grillage or foundation costs. Further first cost savings result from:

1. Reduction in condenser water pump size and power input because of the lower water quantities accompanying higher ranges
2. Reduction in piping costs with lower water quantities.
3. Reduction in tower fan power input.

At a given wet-bulb temperature, an increased condensing temperature can be obtained by one or both of two methods:

1. Increasing the condenser entering water temperature and, therefore, the approach
2. Reducing the condenser water quantity.

With no change in the refrigeration equipment, a higher condensing temperature does result in a higher compressor power input requirement and, therefore, higher operating costs and motor and drive first costs. This effect can be countered by accepting a larger condenser size. With increasing condensing temperature, installed cooling tower costs usually fall more rapidly than refrigeration first costs rise.

Although the compressor full load power requirements are greater with increased condensing temperatures, the compressor motor is only partially loaded most of the time. Power savings from the smaller condenser water pump and tower fan motors which run continuously may more than pay for increased compressor operating costs.

Codes and manufacturers' recommendations should be referred to for limitation on maximum condensing temperatures

Increasing condenser water temperatures may necessitate a more complete and thorough water treatment program than required at lower temperatures.

Atmospheric Corrections

At elevations above sea level, the reduction in weight flow of air due to the decrease in density of the air at high altitude is offset by the greater latent heat absorbing capacity of the high altitude air Therefore, no correction need be made to cooling tower ratings as a result of altitude effects. Fan motor selections made on a sea level basis are conservative.

Fan Drives

Cooling towers are usually obtainable with either gear or belt drives. Gear drives are recommended on large cooling towers since malfunctions are infrequent.

CONDENSER WATER TEMPERATURE CONTROL

Operation of refrigeration equipment at low outdoor temperatures necessitates a control of condensing pressure. Maintenance of a minimum condensing pressure insures proper operation of the thermostatic expansion valve or the refrigerant float valves. A fall in evaporator temperature to the safety control setting is also prevented.

When water-cooled condensing is used, condensing pressure control is achieved thru condenser water temperature control. It is usually not advisable to operate a constant speed centrifugal refrigeration machine at condensing temperatures below 80 F. Variable speed centrifugal machines may be operated at condensing temperatures as low as 50 F Condensing temperatures required for expansion valve operation for normal reciprocating equipment should be maintained at about 90 F

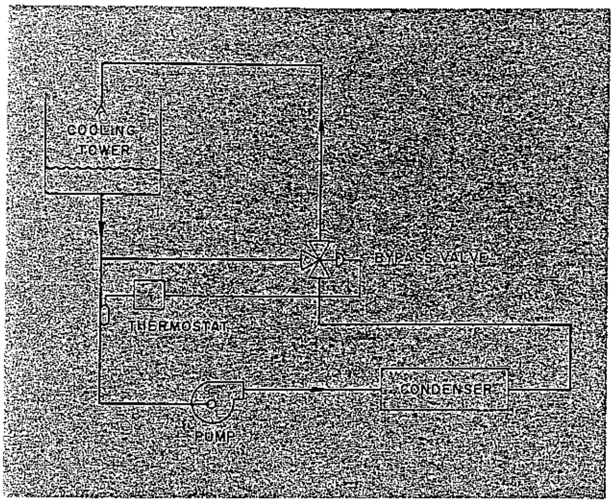


FIG. 53 — CONDENSER WATER TEMPERATURE CONTROL

One or more of the following methods may be used for controlling condenser entering water temperatures:

1. Cycle the cooling tower fan
2. Employ a two-speed fan motor to permit a capacity reduction.
3. Stop in sequence the fans on a multi-cell tower.
4. Bypass the cooling tower fill thru a control valve (*Fig. 53*).

Each of the above solutions may be manually or automatically controlled. The control valve may be snap-acting or may modulate the water bypassed, but should be snap-acting at subfreezing temperatures.

WINTER OPERATION

Winter operation of a cooling tower introduces the problems of water freeze-up in the basin and ice formation on the fan blades and louvers.

The use of a depressed or auxiliary sump within the heated space is one solution to the freezing of water in the basin (*Fig. 54*) In this way, the tower basin is dry during periods of shutdown and whenever condenser water bypasses the fill. This auxiliary sump should be sized to provide sufficient storage space for all the water in the tower and basin and to provide a suction head on the pump

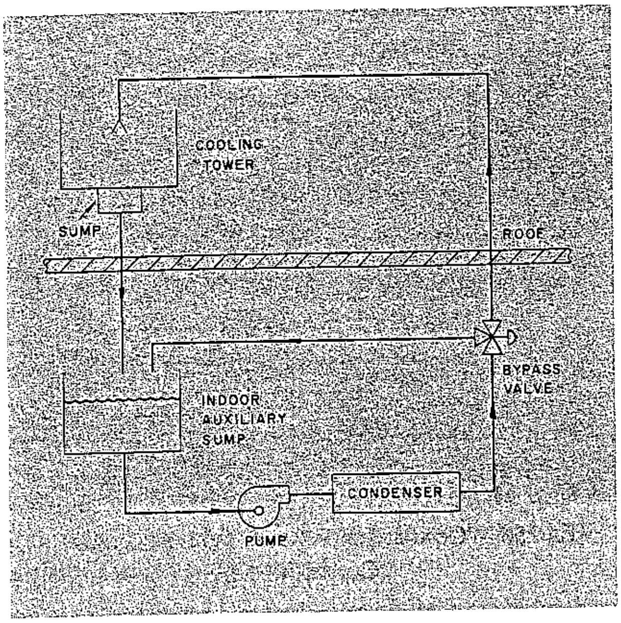


Fig 54 – Condenser Water Temperature Control, Auxiliary Sump

when the tower is operating. An alternate solution provides for heating the basin water with steam or hot brine passing thru coils.

Ice formation on the fan blades results in excessive vibration which may lead to blade breakage and damage to the tower. Two-speed operation may be employed at tower leaving air temperatures approaching 32 F. If ice continues to form at low speed operation, a vibration switch may be utilized to stop the fan.

Air flow thru the tower can be restricted by an ice build-up on the louvers. Complete prevention of ice formation may be difficult, but small deposits can be melted by reversing the fan motor and operating at top speed. This reversing is usually accomplished manually.

MAKE-UP WATER

A cooling tower loses water by evaporation, drift and bleed-off. Evaporation amounts to about one percent of the total condenser water circulated for each 10 degrees of range. Drift loss is constant at all ranges and is approximately 0.2% of the water circulated. Bleed-off varies with water conditions and should be established by the water treatment program as explained in *Part 5*. Where water conditions are not severe, bleed-off is approximately 0.3% for each 10 degrees of range.

The amount of make-up water required is established by the total of these losses. Water may be made up on demand thru a mechanical float-operated valve or by a pair of electric level probes used with a relay and a water solenoid valve.

LOCATION

The selection of a tower site and the orientation of the tower should be guided by the following considerations:

1. Locate the tower so that air circulates and diffuses freely and rapidly without recirculating. Manufacturers usually publish recommendations on this subject.
2. Locate the tower away from sources of heat or contaminated air such as smoke stacks.
3. Locate the tower away from areas where wetting or freezing from normal drift is objectionable. Drift may be carried several hundred feet downwind from the tower if the wind is strong.
4. Locate the tower above or remote from occupied spaces or surroundings with low ambient sound levels. The ideal location is on the roof of a tall building. A tower location at a low level between tall buildings is very undesirable with regard to both tower performance and sound level.
5. Condenser water piping may be simplified and its cost reduced if the tower is located immediately adjacent to or above the refrigeration room.
6. Locating a tower below the refrigeration equipment may lead to problems of siphoning or overflow at shutdown. If water is siphoned from the condenser, the condenser may be damaged by water shock when the pump is started again. If such a location is planned, consideration should be given to reliable checking of backflow at shutdown and to breaking any siphon which might form.
7. Where the tower is to be mounted on a roof, the ability of the roof to withstand the added weight should be checked. The tower should be located to afford the most even distribution of weight on the structural members.

LAYOUT

If the tower is remote from the refrigerating machine or at a lower elevation, the condenser water pump may be located adjacent to the tower. With a tower at grade level, a vertical pump may be used instead of the more conventional horizontal pump.

A steel or wood cooling tower mounted on a roof is elevated and supported by a steel grillage. A tower located on the ground may be so mounted, or may be furnished without a basin and mounted on a concrete basin Grillages and concrete basins should be designed in accordance with manufacturers' recommendations. The design of a concrete basin should include suction screens. Wood and steel basins usually feature sump screens.

On some towers the manufacturers offer a vertical supply line to the water distribution system located at the center of the cell. For certain layouts, this feature can improve the appearance of the installation and reduce the complexity and cost of the piping.

For details of condenser and cooling tower piping, refer to *Part 3, Chapter 2* The following are further specific recommendations.

1. Tower overflows should be piped to a drain and should not be valved.
2. Provision should be made for draining the tower and equipment.
3. A city water hose bib should be provided at the tower to facilitate cleaning.
4. A tower should be provided with a city water fill line in addition to a make-up line A tower located above the refrigeration equipment affords an ideal location for filling the entire condenser water system.
5. If the fan motor starter is remote from the tower, a disconnect switch should be provided at the tower to insure safety while servicing.

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